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A complete index of all Journal technical articles, from January through December, will appear in the December issue. All Journal technical articles are indexed by Engineering Index, Inc. SAE Journal is available on microfilm from University Microfilms, Ann Arbor, Mich.

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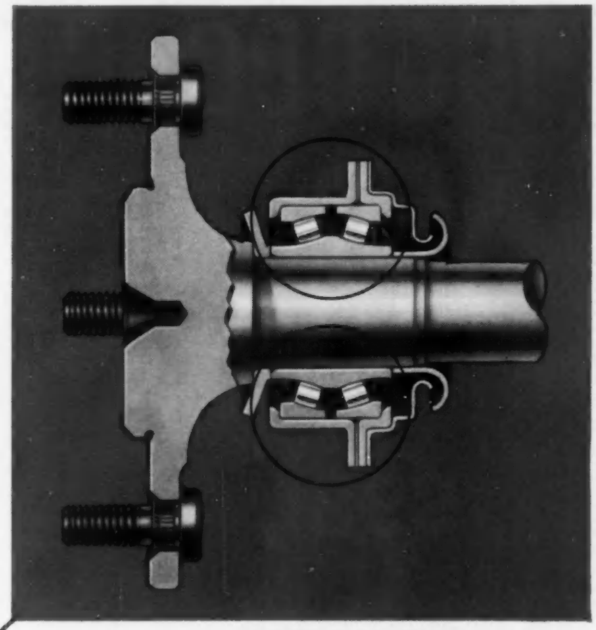

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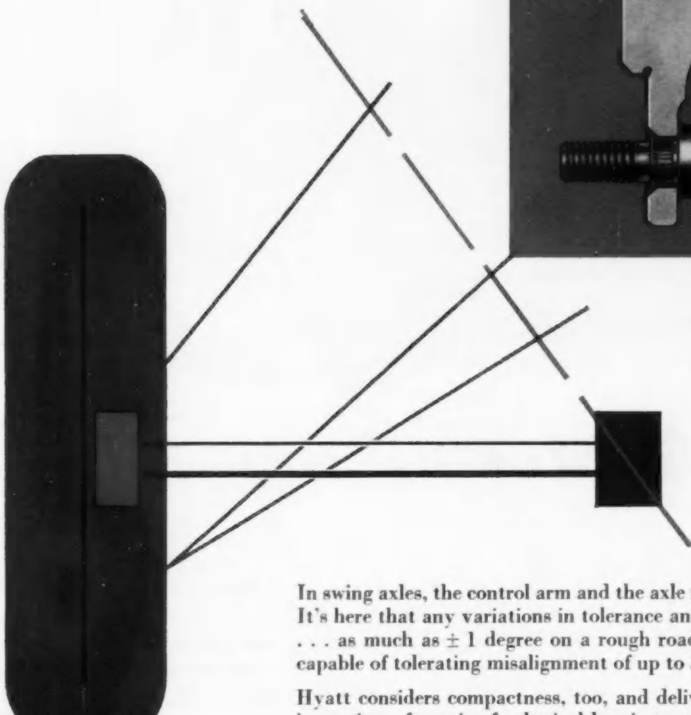

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FUELS & LUBRICANTS

Are Octane Numbers and Hydrocarbon Type Enough? F. D. BUERSTETTA, I. A. CAPUTO, E. S. CORNER, T. M. KORN, E. C. YOUNGHOUSE. Paper No. 206A. Study made by Esso Research & Eng. Co. and Ethyl Corp. to determine whether hydrocarbon-type effect observed in road antiknock studies of gasolines is independent of other properties over and above laboratory octane numbers; results show that road superiority of aromatic over olefinic fuel at moderate-to-high engine speeds varies with other compositional factors; advantages for high motor octane number, high TEL content, and high aromatic concentration indicated; tables. 24 refs.

Tetramethyl Lead — Antiknock for Better Road Performance, R. H. PERRY, JR., C. J. DIPERNA, D. P. HEATH. Paper No. 207A. Study made by Socony Mobil Oil Co., on road octane number depreciation of catalytic reformate gasolines; findings show that tetramethyl lead (TML) is most effective in minimizing depreciation; substitution of TML for TEL increases its road octane number performance in manual transmission cars by virtue of its superior distribution characteristics and in manual and automatic transmission cars by virtue its higher intrinsic antiknock value in aromatic type stocks.

Antiknock Behavior of Alkyl Lead Compounds, H. E. HESSELBERG, J. R. HOWARD. Paper No. 207B. History of Ethyl's evaluation of alkyl lead compounds; changes in engine environment and fuel composition having influence on antiknock effectiveness of various alkly lead compounds as related to TEL; relationships between lead alkyl effectiveness and fuel composition; compared with TEL, tetramethyl lead shows increased effectiveness with increasing aromatic content, leaded octane number level, and lead content, and with decreasing fuel sulphur content and TEL susceptibility.

Utility of Tetramethyl Lead in Gasoline, D. L. PASTELL, W. E. MORRIS. Paper No. 207C. Study by E. I. du Pont de Nemours on antiknock effectiveness of tetramethyl lead (TML) relative to TEL in 75 different fuels; data show that TML has maximum advantage over TEL in cars with poor fuel distribution and in fuels with poor octane distribution over boiling range; TML gives greatest octane improvement by Modified Uniontown method and least improvement by Research method; TML has no adverse effect on gasoline stability; tables.

Tetramethyl Lead Reduces Low Speed Knock, T. M. KORN, G. MOSS. Paper No. 207D. Study by Esso Research Ltd., England, shows that substitution of TML for TEL in fuels having overhead fractions of low octane quality results in improvement in road antiknock behavior; octane distribution is characterized by variable of fuel called "Delta R" found to predict observed low speed advantage for TML, which is more pronounced in European than in American cars; this is probably reflection of increased tendency toward fuel segregation by cars with manual transmissions.

GROUND VEHICLES

Influence of Road Surfaces and Their Variables on Passenger Car Skidding, O. K. NORMANN. Paper No. 205C. Influencing factors are type of aggregate, cement or binder, age of surface, maintenance, foreign material on surface, and climatic effects; data showing variation in friction coefficients at various places on same surface in 108 tests conducted by towing 4-wheeled vehicle and measuring pull

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Briefs of SAE PAPERS

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required and effect of speed on friction coefficients; polishing effect of aggregates; treatment of slippery pavements.

Automotive Sprag Clutches — Design and Application, E. A. FERRIS. Paper No. 208A. Experience gained at Borg-Warner Corp., emphasizing design points for reliable sprag clutch operation in automotive, aircraft, truck, and farm equipment applications with respect to full torque capacity and long overrunning life; method for calculating theoretical torque capacity of clutch; factors which cause dephasing; full phasing in Borg Warner clutch is achieved by use of two cages held concentric by their respective races; concentricity, proportions, material and hardness of races.

Roller Clutch Design, R. E. SAUZEDDE. Paper No. 208B. Review of roller clutch types of 1-way clutches and general method of design consistent with accepted practice by Automotive Industry; each design should be analyzed as to operational requirements on clutch such as shock loading, deflections under load, eccentricities due to manufacturing tolerances effecting usable cam rise, centrifugal effect on rollers, energizing force, etc; recommended design parameters and methods of calculation.

Transmission Gear Design for Strength, E. L. JONES. Paper No. 208C. Considerations involved in design of automotive transmission gears for durability; principal modes of gear tooth failure possible are bending fatigue, compressive fatigue, and scoring; transmission gear materials such as through hardening grade of alloy steel AISI 5132, 5140, or 5145, carbo-nitrided to produce hard case of 0.005 to 0.012 in. depth; carburizing grades of steel such as AISI 4023, 4027, 8620; stress formulas presented are empirical.

Manufacturing Considerations Affecting Transmission Gear Design, A. HARDY. Paper No. 208D. Paper shows how problem of tolerances in gear manufacture affects production engineering, manufacturing and inspection department and why certain compromises are necessary; gear must be designed to accommodate roughing tool such as hob, shaper cutter, or broach; if gear is of type which requires finishing, design should include clearance for rotary shaving cutter, rack shaver, or gear grinding wheel.

Gear Design for Noise Reduction, W. D. ROUTE. Paper No. 208E. Outline of features of design which have bearing on noise sensitivity of train and seven accepted design rules to control gear noise as follows: total contact ratio greater than 2.5, with least involute contact ratio not less than 1.2; to select finest pitch and lowest pressure angle consistent with strength requirements; to avoid tooth action close to base circle, etc.

Motor Vehicle Aspect of Traffic Safety, C. W. PRISK. Paper No. 209A. Study of highway safety conditions throughout United States to determine what Federal action might be taken was performed by Bureau of Public Roads; accident reports show vehicle defects as minor factor; analysis of special studies indicates that useful findings could be developed for vehicle factors; plan consisted of classifying 3 billion vehicle miles of travel exposure to accident by sampling vehicles and drivers on selected sections of main rural highway; findings on major vehicle factors.

California Motor Vehicle Emission Standards, G. C. HASS. Paper No. 210A. Emphasis is on legal framework of motor vehicle exhaust emission standards adopted by California Board of Public Health, Dec. 4, 1959; viewpoint on smog and air pollution which underlies standards, and some of problems involved in their application; in establishing standards for air quality, three levels of concern were established and defined as adverse, serious, and emergency level; latter two formed basis for standards.

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V.I. IMPROVER RESEARCH SHIFTS INTO HIGH. Although viscosity index improvers are relative newcomers, they're doing a man-sized job in many lubricating oils. Continuous research on them is being carried out at Dow—in fact, it's currently being speeded up in Dow's Automotive Chemicals Laboratories. Why? Because the tasks V.I. improvers are asked to perform have rapidly multiplied. Present "extra-curricular" jobs as pour point depressants and detergents are varied and complex; yet vastly improved properties will be expected of the V.I. improvers of a few years from now.

As an example of the specialized, modern research equipment utilized in this work, Dow has developed an apparatus that simulates 20,000 miles of road testing in just one hour's time! Similar testing devices are now being perfected in the lab.

RESEARCH PAYS OFF IN THE BOILER ROOM. Newcomers to the industrial scene, Dow research on chelating agents led to their application as boiler scale removal agents. These special chemical "claws" grab and hold metal ions in solution, rendering them inactive. Versene®, one of Dow's three chelating agents, speeds descaling many times over older standard methods. Other high-quality, low-cost members of the Dow family of chelating agents are Versenol® and Versenex*. *TRADEMARK

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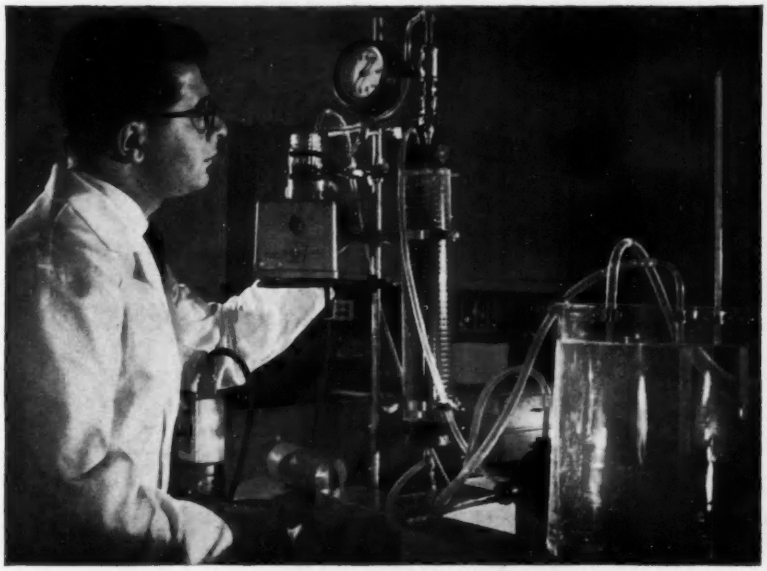
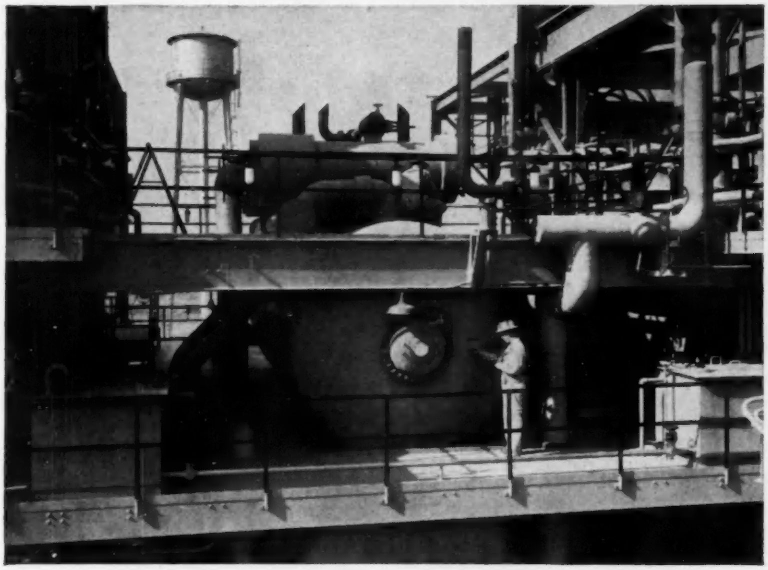


Shear-testing V.I. improvers under "road test" conditions.



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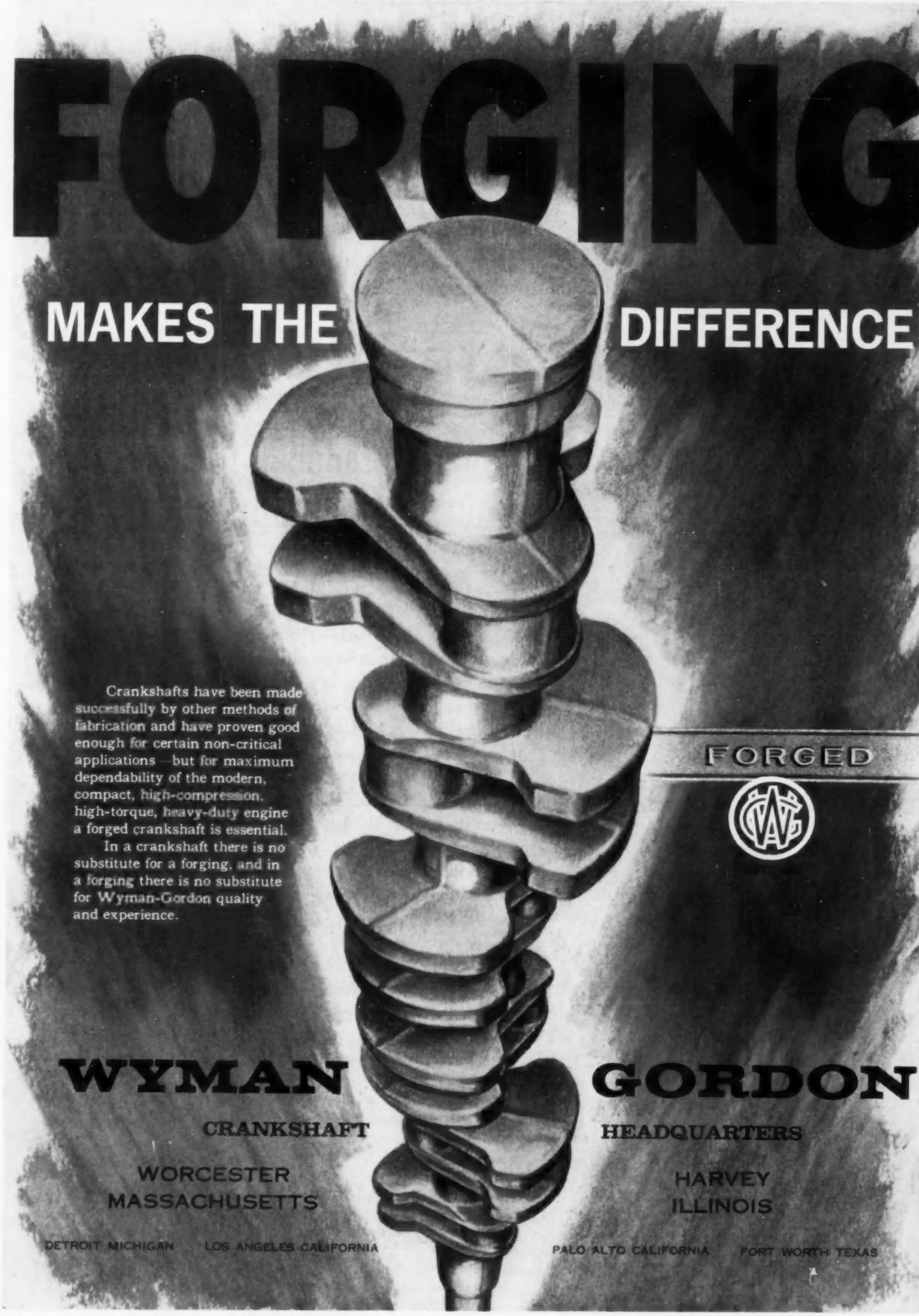
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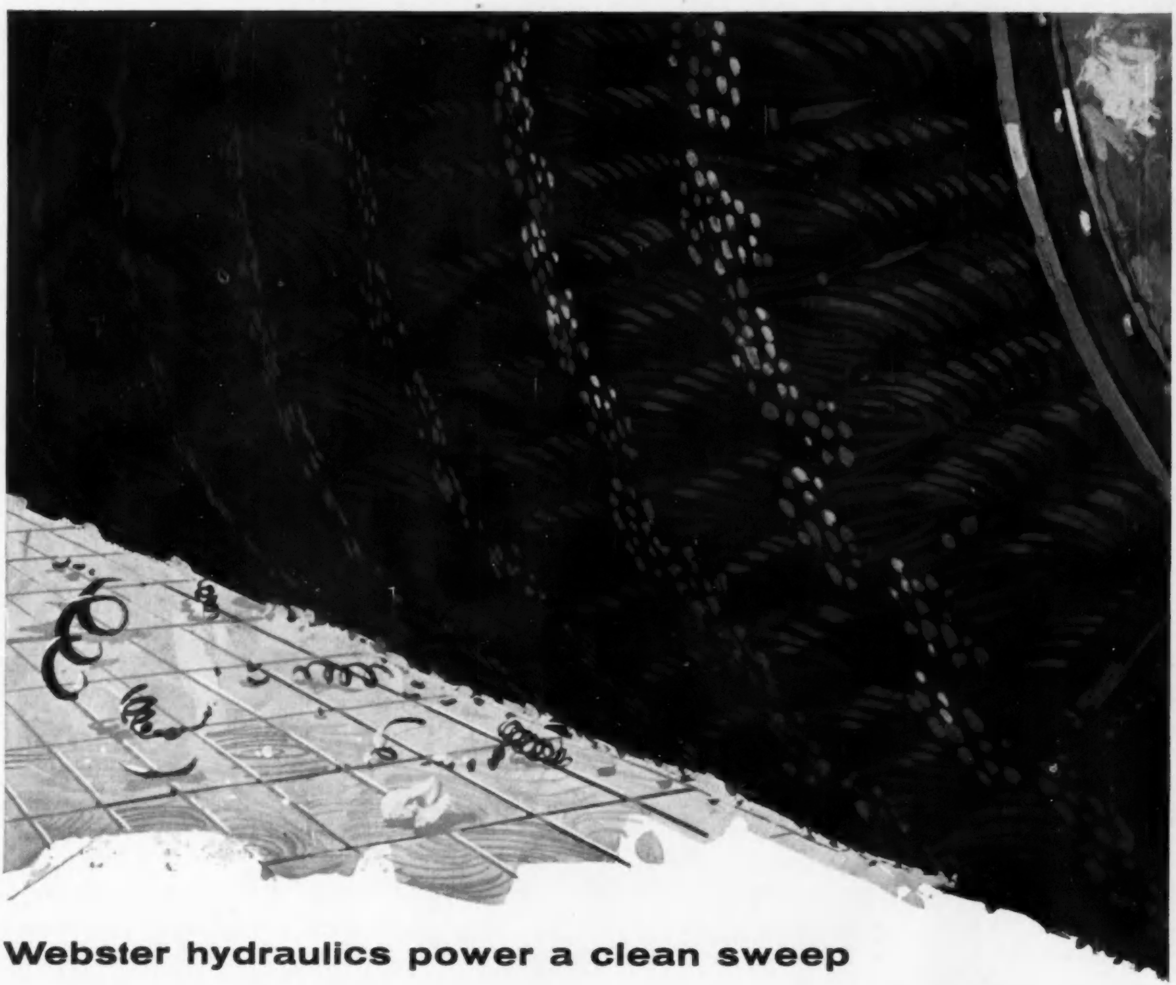

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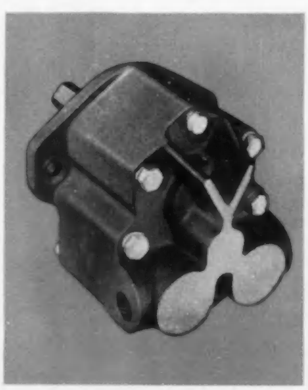

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For Sake of Argument

A Committee Can be a Team . . .

COMMITTEES are sometimes called teams . . . and maybe when a committee thinks of itself as a team, it makes surer progress.

When a committee bogs down, it usually has lacked clear-headed, aggressive leadership and/or a brief, understandable, focus-in-writing of its objective. It rarely has the focus without the leadership, because provision and maintenance of focus are a leader's main job.

Now, effective leadership is an accepted requirement for a team. But it is often resisted by members of a committee.

A team is automatically thought of as a group whose aim has been established (to get more runs, to score more points, or to win an objective). A team, in short, *expects* to accomplish something specific.

The leader of a team is expected to select the team members, give them assignments, and help and urge them to successful completion of accepted responsibilities. A leader who leads is a *sine qua non* of every team. No winning team ever went on the field under the direction of a moderator.

William J. Schleicher decries the use of committees as creative planning groups or initiators because, he feels, they tend to "replace brilliant individual thinking with low-level mumbling." In a penetrating, 450-word editorial in *Carbide Engineering* for April, 1960, Schleicher says:

"I do censure the philosophy which permits committees to sit in judgement on the bold schemes of leaders. . . . Leaders have always been scarce, and are becoming scarcer. Then why muzzle those we have with a committee?"

Schleicher's question poses a dilemma, of course, because committees will probably continue to be the means on which many businesses and all technical societies rely to get things done.

The committee that thinks of itself as a team is less likely to merit Schleicher's censure than the one that works under the aegis of a moderator.

Norman G. Shidle

chips

from SAE meetings, members, and committees

FIRST LOW-PRESSURE PROCESS CASTINGS ever produced probably were for the Model T Ford in the '20s, according to Aluminum Co.'s Marius Guyot, veteran of 46 years of association with the permanent mold process.

"It is interesting to note," he told an SAE production group recently, "that the low-pressure process was used at the Ford Motor Co. in the '20s for producing small castings for the Model T.

"The setup was a holding furnace with a pressure-tight cover. Air pressure of 1 psi was supplied to this furnace through a pipe. A pipe with a bleed-off hole for the air was located convenient to the operator so that the furnace was not pressurized at all times. . . . To produce the casting, the operator had only to place his thumb over the bleed-off hole for 5–10 sec, thus pressuring the furnace. Removing his thumb released the pressure.

"The metal in the sprue returned to the furnace. The mold was then opened and the casting removed.

"The process then dropped out of foundry recognition until recently. The Ford setup was as simple as the Model T!"

IS MAN A UNIQUE form of life? Is there elsewhere in the universe "intelligent life capable of recognition as life, and capable of some form of communication?

Having explored the evidence Col. Paul A. Campbell, assistant to the Commander (Advanced Studies), USAF, Medical Corps, draws some interesting conclusions. He says:

"The proper timing of the proper events, in the proper sequences in the proper environmental circumstances; the proper interaction of all in the proper statistical framework; along with multitudinous other factors has resulted, in my opinion, in unique man.

"There is probably much other life in the universe, if one defines life as macromolecules capable of replication. Possibly there is life based on silicon rather than carbon. Possibly there is life steeped in H_2S or ammonia, or various other substances. But would it be recognizable as a life with which we might communicate or associate? Could we learn anything from it if we could not sample and study it?"

GRAVEL IN RUNWAY SLUSH does much more harm than just slush alone. British-European Airways had to change three engines on one of its jet transports as a result of the transport's taxiing through a gravel-slush mixture at Copenhagen recently.

•

EUROPEAN AIRCRAFT BUILDERS are working on a swing-nose cargo transport.

MOTOR VEHICLES TODAY are traveling an average of 16 mph faster on our most modern highways than in 1941, Bureau of Public Roads studies indicate. . . . In 1941, it was rare to find a location where the average speed approached 50 mph. Now, the average speed is around 64 mph; and 25% of the vehicles exceed 70 mph.

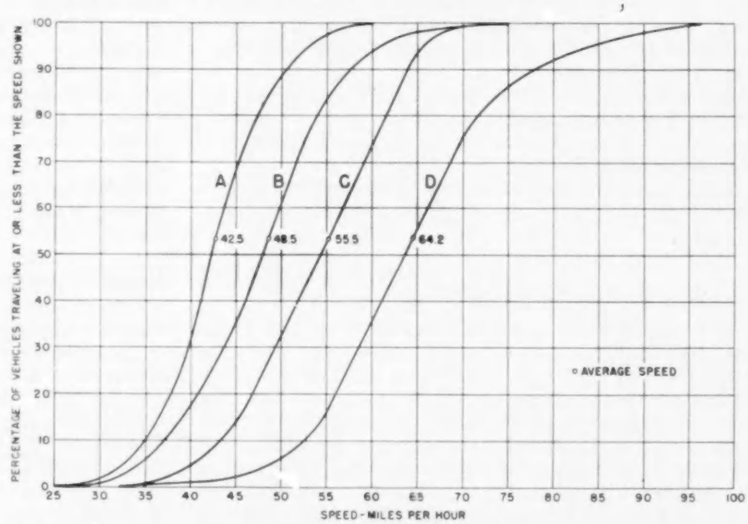


Fig. 1 — Distribution of normal passenger-car speeds. Speed distributions on most main rural highways in 1941 would have fallen between curves A and B. Where a 60 mph speed limit is well enforced, speeds are as shown by curve C which corresponds with present speeds during light volumes on the New Jersey turnpike. Today's highest speed distributions are represented by curve D. These speeds occur on rural sections of modern freeways where there is no speed limit or where there is no enforcement of an existing limit.

What do we *really* know about conditions

Based on paper by
E. N. Parker
Enrico Fermi Institute for Nuclear Studies, University of Chicago

PRESENT KNOWLEDGE of conditions in interplanetary space is largely qualitative and conjectural. A significant portion of comments based on current knowledge may soon be rendered obsolete by direct observation of interplanetary conditions from instruments in space.

Most of our present knowledge is based on indirect inference. It is clear, for instance, that interplanetary space is a remarkably energetic region of continually interacting magnetic fields and plasmas . . . and that the more violent interplanetary upheavals follow as the result of "sunspot activity" (solar activity) on the sun. "Solar flare" — observed as an enormous brightening of atomic hydrogen line emission in the chromosphere — is the principal indication of the kind of solar outburst leading to interplanetary disturbances.

Solar flare is observed a few thousand kilometers above the visible "surface" of the sun (the photosphere).

Observational inferences

The visible flare is accompanied by strong emission (up to 10^{22} ergs) of radio noise from the solar corona above the flare site. When a flare is seen in profile at the edge of the solar disc, one observes luminous regions squirting up and out of the flare.

Most of the larger flares apparently generate large quantities of high-speed protons with energies of 10–100 Mev, and on rare occasions up to many thousands of Mev. The protons soon fill a large portion of interplanetary space and are observed at Earth. The 10–100-Mev solar flare protons are observed at Earth an hour or so after the flare, as a consequence of the extra ionization the protons produce in the terrestrial ionosphere.

The protons have also been observed in particle counters carried by balloons to the top of the atmosphere at high geomagnetic latitudes.

Now so far as the general interplanetary upheaval following the solar flare is concerned, we must state some of the conditions prevailing in the region of the flare.

In the solar corona the temperature is normally between one and two million degrees absolute, the temperature being maintained, presumably, by the mechanical dissipation of acoustical and hydromagnetic waves generated beneath the photosphere.

Solar material is blasted out into interplanetary space, perhaps by hydrodynamic means. One or two days later we find at Earth brilliant auroral displays at relatively low latitudes. There is then a geomagnetic storm, in which at low latitudes the mean magnetic field suddenly increases by a fraction of a per cent for a few hours, and then decreases by a per cent or more. Locally, the field undergoes violent, irregular oscillations of a couple of per cent until after 10 hr or so the field quiets down and begins to relax back up to normal.

The cosmic ray intensity at Earth usually decreases during the geomagnetic storm by amounts running up to 30 or 40% This phenomenon is called the Forbush-type cosmic ray decrease, after its discoverer. The particle intensity in the outer Van Allen belt declines somewhat during the magnetic storm, and then for some unaccountable reason rises to extraordinarily high levels once the geomagnetic disturbances have quieted down.

All of these terrestrial phenomena have been attributed to the streaming of a cloud of ionized hydrogen past Earth. The solar flare is the signal that such a cloud will be ejected from the sun. The one- and two-day delay for arrival at Earth (at a distance of 15×10^{13} cm) indicates a velocity of 1000–2000 km/sec, for which 1500 km/sec is a standard medium figure. The density of the plasma cloud has been estimated to be perhaps as high as 10^5 hydrogen ions per cm^3 at the orbit of Earth on some occasions, although such estimates are not much more than rough guesses based on the consequences of the passage of the cloud.

Extensive calculations have been able to show that the geomagnetic storm is not an implausible consequence of the impact of the plasma cloud

in interplanetary space?

against the geomagnetic field.

It has been shown, too, that the impact of the cloud can plausibly lead to the injection of electrons and protons of kilovolt energies down along the geomagnetic lines of force leading to the middle latitudes, where aurorae are observed during geomagnetic storms. . . . And it is generally assumed that it is just such particles that are responsible for the aurora.

Several ideas have been proposed as to how the cloud may sweep back the cosmic rays from the inner solar system to produce the Forbush decrease, though none of the ideas to be found in the literature up to now have been able to withstand the two tests of theoretical investigation and the expanding quantitative observational knowledge of the Forbush decrease phenomenon.

Altogether, then, the idea that a plasma cloud is accelerated outward from the sun at the time of a large flare, originally suggested by Lindemann three decades ago, now has a broad basis in our knowledge of the abnormal terrestrial phenomena observed a day or so later. No other hypothesis besides solar corpuscular emission — that is, an ejected plasma cloud — can begin to explain the simultaneous storm, aurorae, Forbush decrease, and other phenomena. We give 1500 km/sec as the cloud velocity. The density of the cloud is a much more nebulous figure. We suggest 10^3–10^5/cm^3 as necessary to explain the terrestrial consequences of the passage of the cloud. The size of the cloud can be estimated from the angular width of its zone of influence outward from the site of the flare, and we arrive at 90 deg as its nominal width (2×10^{13} cm at Earth). The cloud must carry magnetic fields because of its cosmic ray effects, and their magnitude has recently been measured at a few times 10^{-4} gauss. But we do not know anything directly of the configuration of the lines of force, nor their direction.

This then is our meager picture of the plasma cloud, which comes blasting out from the sun when there is a solar flare.

What is, in fact, the ambient state in interplanetary space, when a plasma cloud is not blasting outward at 1500 km/sec from a flare?

What are conditions during the years of minimum solar activity when for periods of several months there may be no outburst on the sun?

We might presume that the ambient state during solar activity is a disrupted form of the ambient state during quiet periods without activity.

Some qualitative conclusions about interplanetary magnetic fields can be gleaned from cosmic ray observations.

We observe, for instance, that 10–100 Mev solar protons arrive at Earth 1–6 hr after a flare on the visible atmosphere of the sun, and may be detected for a day thereafter. Rarely, if ever, do we see protons which are not attributable to a flare on the visible hemisphere, and hence might have been attributed to a flare on the back side of the sun.

From these facts, we deduce that:

1. High-energy solar protons are trapped in the inner solar system for periods of up to a day.
2. Any interplanetary fields in excess of 10^{-6} gauss must be radial to the extent that protons, constrained to move principally along the lines of force, can get to Earth in times only a few times the straight-line propagation time.
3. There must be fields which do not allow particles from the back side of the sun to arrive at Earth.

There are reasons for believing that the normal galactic cosmic ray intensity is the high level observed at sunspot minimum when the sun is quiet for long periods. Thus we conclude that continuing solar activity must in some way sweep back the galactic cosmic rays for a period of several years, perhaps in the same way that the plasma cloud from a single solar flare may sweep back the cosmic rays for a few days to produce the Forbush decrease. Recent observations from Pioneer V several million kilometers toward the sun from the orbit of Earth show the same cosmic ray depression as at Earth. Thus we can rule out some of the older ideas that the depression is due to a magnetic shielding of the

conditions in interplanetary space

... continued

inner solar system by, for instance, a solar magnetic dipole field, for then we would observe a significant gradient in the cosmic ray depression over the millions of kilometers separating Earth and Pioneer V. The very close equality of the cosmic ray intensity at these two widely separated positions argues for a largely radial field if there is any significant outward convective motion at all.

On the other hand, in apparent conflict with these conclusions is the recent magnetic data of Coleman, Davis, and Sonnet (1960), suggesting a quiet-day 2×10^{-5} gauss field *perpendicular* to the plane of the ecliptic. As they point out, their data cannot be reconciled with our other information, nor can it be easily ignored. We must accept any argument with caution.

Theoretical inferences

The theoretical problem has been attacked from two points of view.

One view begins with the premise that the interplanetary medium is static and is simply the normal hydrostatic extension of the solar corona.

The other is that the interplanetary medium is an extension of the solar corona which is expanding outward at several hundred kilometers per second.

There are, of course, many additional theoretical ideas and arguments that have been proposed.

Gold (1959, 1960) has suggested that the plasma cloud from a flare may carry with it the outer portions of the sunspot fields in the vicinity of the flare, producing a "magnetic tongue" extending ultimately past the orbit of Earth. The tongue is then a magnetic channel which excludes galactic cosmic rays, to produce a Forbush decrease throughout its interior and to store the solar flare protons for the day or two that is observed.

Many more interesting ideas could be mentioned. But with the imminent scheduling of space probes carrying instruments to observe directly the magnetic fields, ionized gases, and cosmic rays in interplanetary space, further discussion before the fact would seem unprofitable.

The final picture will eventually emerge from the observations, and we may expect that several surprises will be in store for us.

This article is based on part of an Astronautic Symposium developed jointly by SAE and the Air Force Office of Scientific Research. The Symposium is available only as a book, titled "Vistas in Astronautics — 1960." To order, turn to p. 6.

MEASURING Early in Design

Measuring reliability is the

This article details

Based on paper by

William L. Johnston

Convair Division, General Dynamics Corp.

THERE was a time when an airplane designer rarely knew the actual weight of an airplane until it was built and weighed.

Today, he knows at any time during the design and development stage exactly how he stands with respect to target weight.

The safety of a design can be similarly estimated for use throughout the design process . . . by measuring reliability, which utilizes the measures of the fields of probability and statistics. These are harder for a designer to understand than are measures like pounds. But with training and assistance of trained personnel, he can learn.

One practical system for measuring safety involves six specific steps, results from the first three of which may be readily recorded in the "Component Failure Analysis" form shown in Fig. 1.

The six steps are as follow:

Step 1. List for each component all possible types of failures.

Step 2. For each failure, examine its results on the system. This step can best be accomplished by

SAFETY

Process


COMPONENT FAILURE ANALYSIS

Component Name ____________

Component Number ____________

System ____________

Failure Mode	Result on System	Type of Failure*

* Maintenance (M), Mission (R), or Safety (S).

best way to measure safety.

a specific method.

utilizing a tool developed in reliability. A schematic of the system is drawn which shows what components are in parallel, standby, or in serious configuration from a safety standpoint.

Step 3. Classify failure as maintenance, mission, or safety type.

Step 4. For each component, analyze the failure which would affect safety and estimate a failure rate for this type of failure. The estimate can be made by either analyzing the failure records of the reliability group or, if the component is a new design, obtain assistance from reliability engineers to apportion the component failure rate.

The failure records of a reliability group should give not only component failure rates but also should provide a failure rate by type of failure. The failure rate of a component (λ_T) is the sum of its maintenance failure rate (λ_M), its mission failure rate (λ_R), and its safety failure rate (λ_S). This relation can be expressed $\lambda_T = \lambda_M + \lambda_R + \lambda_S$. For many components, the safety failure rate would be zero depending upon the type of component and how it is utilized in the system.

Step 5. The probability of a component not failing in a manner which will endanger human life (safety) can now be computed. If, for example, it can be assumed that the component has a constant failure rate, then the safety of the component would be given by the expression $S = e^{-\lambda_s t}$, where t is the time or number of cycles for which the probability of no safety failure is desired. If some other failure distribution is assumed, such as the Gaussian, then the appropriate probability function is utilized.

Step 6. After the safety of each component has been computed, these values can be combined by the usual probability theorems to find the safety of a system. These probability theorems would be used in the same manner as when the reliability of a system is analyzed. For example, if two components are in series, from a safety standpoint, the safety of this system would be given by the product of the safety of each component.

The method as described above will enable the designer to compute an estimate of the inherent safety of a system. This design parameter can now be quantitatively traded-off with other design parameters. It can be expected that as the data on component failure rates increase, this method of estimating the safety of a system will become more and more precise.

To Order Paper No. 229D . . .
from which material for this article was drawn, see p. 6.

Short cut to

Fuel economy data

Based on paper by

P. M. Clayton

Ford Motor Co.

SEVEN easy steps give fuel economy data on which decisions about car modifications may be intelligently made. These steps blend emperical data with the major variables affecting economy. At the same time, they ignore the lesser factors which complicate calculations without substantially changing the data being sought.

Step 1

Method

Calculate vehicle power requirements.

Remarks

A method of making this calculation is not detailed, since each engineer has preferred methods such as wind tunnel tests, formulas, or road tests of actual vehicles.

Step 2

Method

Find horsepower and rpm required at flywheel.

Remarks

This is also routine, since driveline efficiencies can be closely approximated by experience. Speed of vehicle versus engine rpm is found from tire size, axle ratio, and transmission characteristics.

Step 3

Method

Find motoring friction horespower at wot from:

$$\text{fph} = 5.2 \times 10^{-6}\ QN + 1.027 \times 10^{-8}\ QN^2$$

where:

$Q = D\ \sqrt{0.75\ S/B}$
D = Displacement, cu in.
S = Stroke, in.
B = Bore, in.
N = Engine speed, rpm

Remarks

This is based on emperical data. An example of the accuracy is:

Engine	Fph Calculated	Fph Tests	Number of Engines Tested
A	19.6	21.7	6
B	35.5	34.0	7
C	41.57	42.4	8
D	43.8	46.5	7
E	52.75	52.0	6
F	53.76	54.3	5

Step 4

Method

Find additional pumping losses for road-load operation from vehicles with similar *power to weight* ratio or from:

$$\text{php} = \frac{PDN}{792{,}000}$$

where:

$$P = 0.491\left[K(1-R) + 0.20\left(\frac{N}{1000}\right)^2 (R^2 - 1)\right]$$

P = Manifold pressure difference, psi

$$R = \frac{\text{Vehicle hp required at flywheel}}{\text{Full throttle “as installed” hp at } N \text{ in question}}$$

$K = 21$ when $D < 200$
$= 23$ when $D = 200$ to 350
$= 25$ when $D = 350$ to 450

Remarks

Since friction hp is found for wot operation, additional pumping loss at road load must be added.

Step 5

Method

Calculate indicated hp from:

ihp = flywheel hp + friction hp at wot + php

Step 6

Method

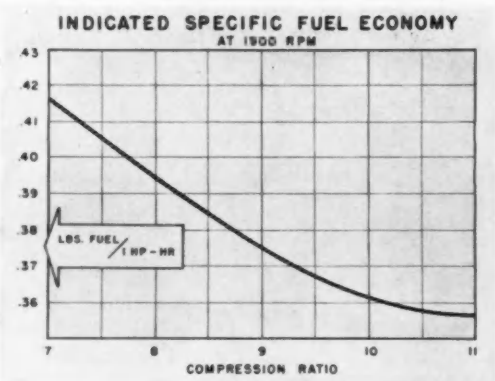
Determine indicated specific fuel consumption by first reading the isfe at 1900 rpm on Chart 1 for the compression ratio of the engine. Then multiply this answer by the correction factor in Chart 2 for the engine speed of interest.

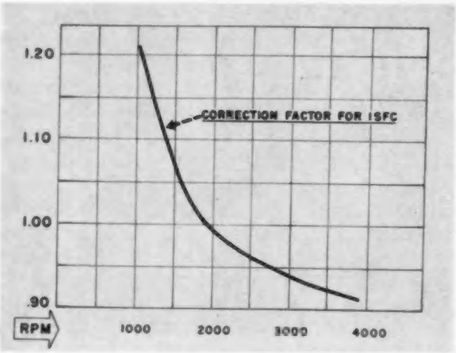
Remarks

The first chart assumes that the indicated fuel consumption of any automotive gasoline engine at a given compression ratio is independent of load and design. Although this isn't true, the numerous other variables are limited enough in their effect to make this calculation useful for evaluating design changes in the vehicle.

The second chart is based on the observation that the indicated thermal efficiency of engines vary in a similar manner with rpm.

(See SAE Paper No. 150A, Several Possible Paths to Improved Part-Load Economy of Spark-Ignition Engines, by A. E. Cleveland and I. N. Bishop.) With this as a start, the curve shown has been adjusted empirically to account for factors such as spark advance modified from mbt, and low rpm carburetor calibration.





See next page for step 7 ⟶

Fuel economy data

. . . continued

Step 7

Method

Find maximum fuel consumption from:

$$\text{mpg} = \frac{\text{mph} \times \text{fuel density}}{\text{ihp} \times \text{isfc}}$$

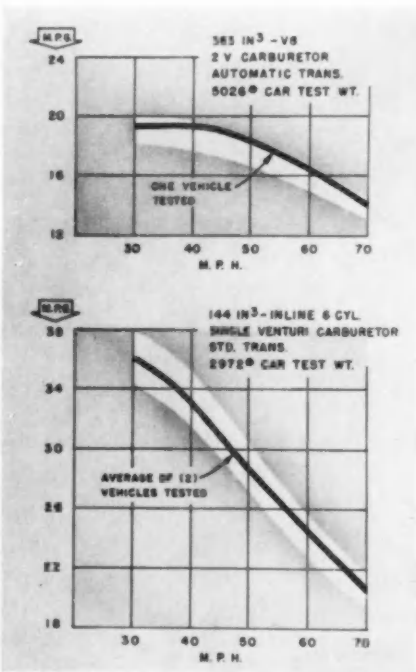
where:

mph = Vehicle speed, mph
isfc = Results of Step 6
ihp = Results of Step 5
and fuel density is measured in lb/gal.

The minimum fuel consumption is 10% less than the maximum value.

Remarks

Calculated results (white band) versus test results (line) are shown for two types of vehicles.



To Order Paper No. 199B . . .
from which material for this article was drawn, see p. 6.

Problems and solutions in

developing Deere's big-scale farming tractor

Based on paper by **F. C. Walters, Merle L. Miller, and Robert H. Tweedy**
John Deere Tractor Research and Engineering Center

A LARGE wheel tractor designed to meet the power needs of big-acreage farming is quite a different vehicle from the ordinary agricultural tractor. In the development of the Deere model 8010 — the 21,000-lb vehicle shown in Fig. 1 — many of the problems uncovered were not to be solved through experience had with smaller, higher-volume farm or industrial tractors.

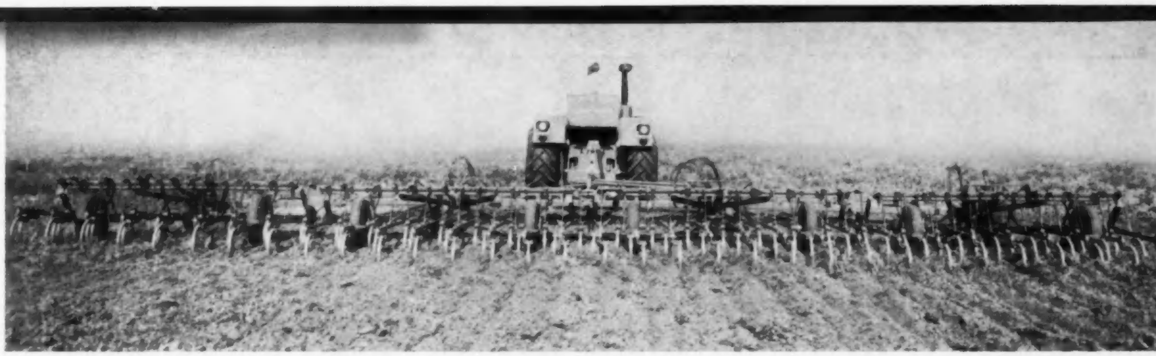
Field test with the prototype proved that a torque converter could not span the wide gear steps of a 4-speed transmission adequately to give good performance. Accordingly, the production model is equipped with a 9-speed transmission with a clutch of our own design. Closely spaced gear reduction ratios in the 2–7-mph travel range were found essential for agricultural operation.

Axles used in similarly powered construction and industrial equipment proved totally inadequate for farm service. This was learned the hard way by having to increase differential capacity twice in order to attain the durability required under the continuous loading and high power factor met on large farms.

Working versus transport loads

Tires with fewer than 12 plies were not strong enough in sidewall or bead strength, or in load-carrying capacity for conventional operations and with integral implement hitch. Experience on smaller tractors with integral implement operation has shown that tire transport loads are higher than field operating loads. With the model 8010, several implements can impose higher working tire loads than transport loads with tools as heavy as the moldboard plow.

Spring cushion seats having a frequency range of 3 cps, which are used on conventional tractors, were



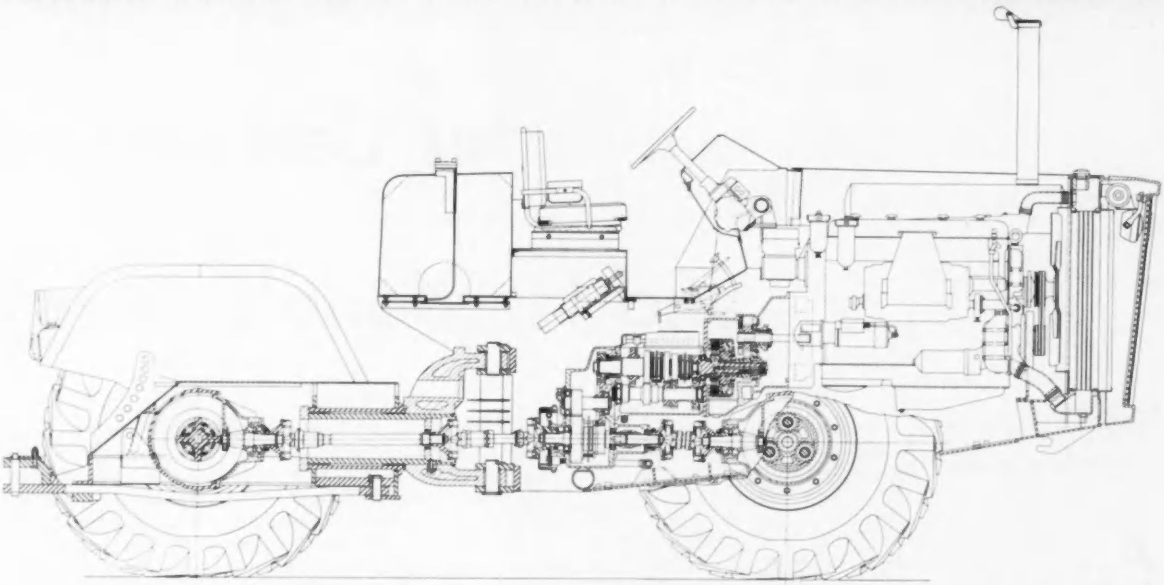


Fig. 1 — Sectional view of the Deere model 8010 tractor designed for large-scale farming operation. Powered by a 2-stroke diesel engine, it features 4-wheel drive and articulated steering.

quite uncomfortable on the 8010. The natural frequency of the 8010 is in the range of 2.6–2.8 cps, thus requiring a seat suspension system that will isolate vibrations down to nominally 1.6 cps. The present cushion has no springs and no more padding and foam rubber than to provide static comfort.

Integral implement hitch

Implement hitch studies demonstrated the desirability of the three-point hitch linkage regardless of tractor size. Equally important on large tractors is the spacing of the link-end of the three points for ease of attachment.

Two 5×16 SAE standard cylinders permit this hitch to raise a 4600-lb, 8-bottom plow to a tail wheel ground clearance in excess of 6 ft in less than 3 sec. It was found best to have the cycle time on this hitch similar to that on smaller tractors rather than the 3–6-sec cycle time in the SAE standard for this size of cylinder. This is because the desired lift time is a function of tractor travel speed.

Experimental work with the hitch disclosed the shock loads imposed on the various members to be from 1.6 to 1.8 times the normal working load. Most severe are the transport loads, "drop and catch" loads, and those imposed when turning with the implement in the ground.

Four-wheel drive

It was decided initially not to provide differential action between front and rear axles. This decision has been proved sound by measurement of torque and power distribution to the axles. Only when the tractor is being transported on firm ground or concrete is there a tendency for negative driveline torque to one of the two axles. And this torque is reduced to a negligible amount if the rolling radius of the front and rear tires is the same. For transport on hard surfaces, a control is provided for disengaging the front wheel drive. In soft ground or tilled fields, there is no problem of torque distribution.

The peculiarities of 4-wheel drive, the effect of tracking wheels, and the matter of dynamic front and rear weight distribution, have yet to be fully explored.

Specifications

Deere Model 8010

Engine	G.M. 6-71E, 2-stroke diesel
Rated speed, rpm	2,100
Drawbar, hp	150 (est.)
Wheelbase, in.	120
Overall width, in.	96
Weight, lb	21,000
Speeds, mph	1.92-17.93

To Order Paper No. 225A . . .
from which material for this article was drawn, see p. 6.

How to Select Proper Case Depth for Carburized and Hardened Gears

Based on paper by

R. Pedersen and S. L. Rice

Caterpillar Tractor Co.

A METHOD has been developed to provide a more rational approach to the selection of case thickness of carburized and hardened steel gears than the use of tooth thickness, which is inadequate for present-day nonstandard gear design. In addition to being a useful design tool, the method aids in determining the cause of failure on gear tooth profiles.

Table 1 — Differences Between Pitting and Case Crushing Failures of Gears

	Pitting	Case Crushing
Physical Appearance	Shallow	Deep, rigid
Occurrence	Gradual	Sudden
General Shape	V-Shape	Longitudinal gouge
Distribution	Many teeth	One or two teeth
Direction of surface cracks	Acute angle to surface	Normal to surface

[1] "Stress Distribution and Strength Condition of Two Rolling Cylinders Pressed Together," E. I. Radzimovsky. University of Ill. Engineering Experiment Station Bulletin, No. 408, 1953.

[2] "Stresses due to Pressure of One Elastic Solid upon Another," by H. R. Thomas and V. A. Hoersch. University of Ill. Engineering Experiment Station Bulletin, No. 212, 1930.

Distinction must first be made between two types of gear tooth profile failure. One type is pitting fatigue; the other is case crushing. The characteristic differences between these two are summarized in Table 1. Case crushing has been found by test to correlate with the thickness of the carburized case. It is believed that such failures are related to the ratio of shear stress to shear strength of the material at the subsurface level where the hardened case meets the softer core.

The method is one of calculating the subsurface shear stresses in a gear tooth and the subsurface shear strength of the material, and from this calculating the maximum value of the shear stress/shear strength ratio (hereafter called the stress/strength ratio).

Calculating subsurface shear stresses

If two mating teeth are assumed to be equivalent to two mating cylinders having curvatures equal to those of the gear teeth at the line of contact, the subsurfaces stresses can be calculated by methods described by Radzimovsky[1] and based on the work of Thomas and Hoersch.[2] When two cylinders are pressed together, the pressure distribution is as shown in Fig. 1. The principal stresses σ_x, σ_y, and σ_z at points in the *XZ* plane and at various depths below the surface can be plotted as in Fig. 2. To generalize the curves, the ordinates are in terms of b, half the width of the band of contact, and the abscissas are in terms of p_{max}, the maximum contact pressure. The shear stress, τ_{45}, on a plane at 45 deg to the *Y* axis and parallel to the *Z* axis, is computed from the relation:

$$\tau_{45} = \frac{\sigma_x - \sigma_y}{2}$$

The values of τ_{45} (for points beneath the surface and in the *XZ* plane) are also plotted in Fig. 2, where it is seen that τ_{45} reaches a maximum value at a depth approximately 0.78b below the surface. The

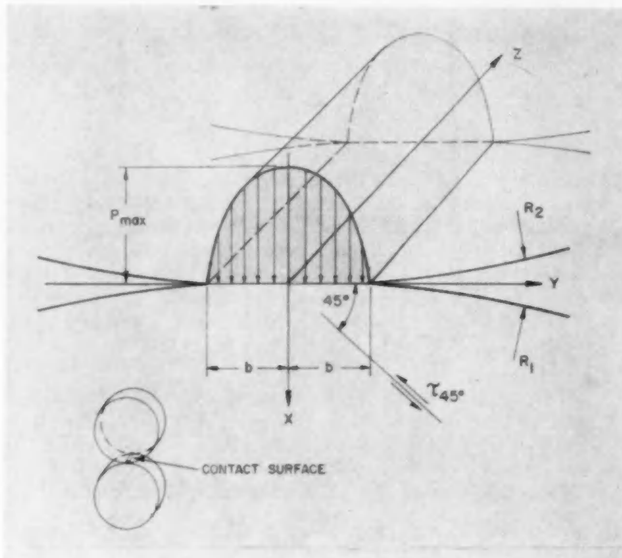
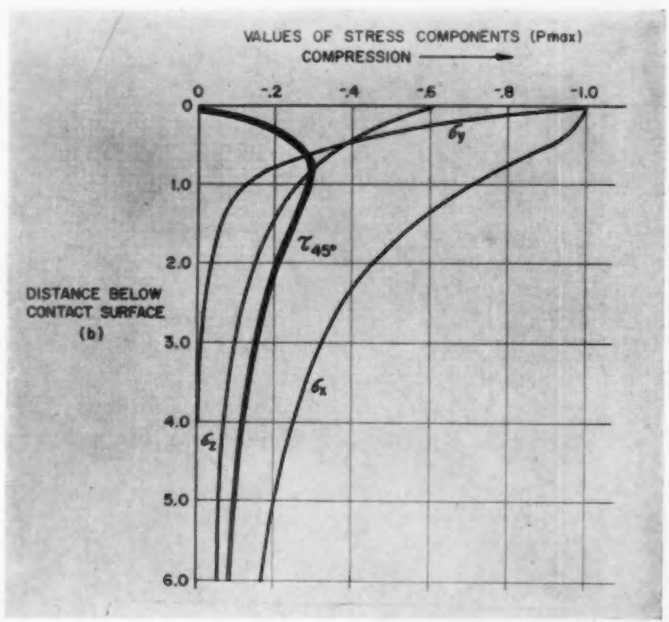


Fig. 1 — Pressure distribution on contact surface of two cylinders.

Fig. 2 — Distribution of principal stresses in a cylinder due to a normal load.



value of τ_{45} at this point is frequently used in applying the maximum shear theory to analyses of pitting failures.

In case-hardened parts there are also residual stresses resulting from heat-treatment. When these residual stresses σ_{xr}, σ_{yr}, and σ_{zr}, in the *X*, *Y*, and *Z* directions, respectively, for a carburized and hardened cylindrical bar are plotted on axes similar to those in Fig. 2, the general shapes of the curves are as shown in Fig. 3. Actual values of the residual stress components can be had from work done by the SAE Iron & Steel Technical Committee, Division IV.

When the principal stresses (Fig. 2) are added to the residual stresses (Fig. 3) the resultant principal stresses will be similar to the curves in Fig. 4 which also shows the shear stress τ'_{45}.

Calculating subsurface shear strength

Cross-section the gear tooth and obtain a hardness gradient curve, similar to the one in Fig. 3, in the region of the line of contact. Several publications provide conversions from hardness to average tensile yield strength, based on the assumption that 90% or more of the material is martensite. With

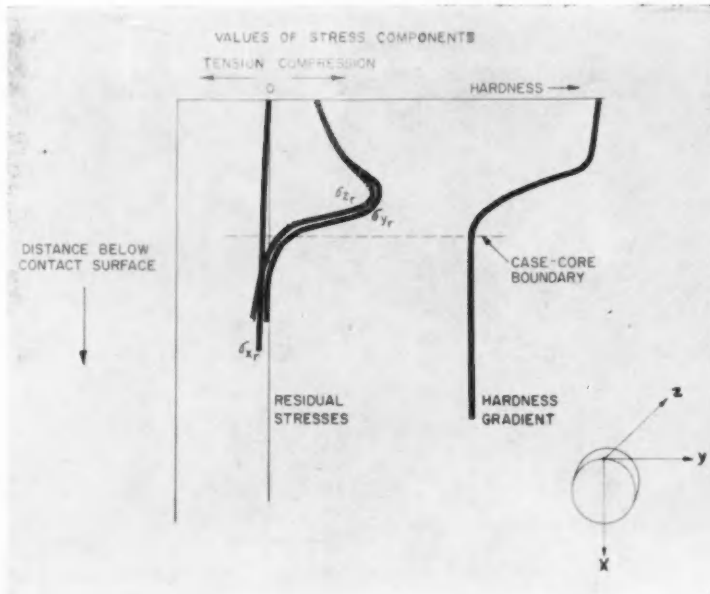


Fig. 3 — Typical residual stress components and hardness gradient for a carburized and hardened cylindrical bar.

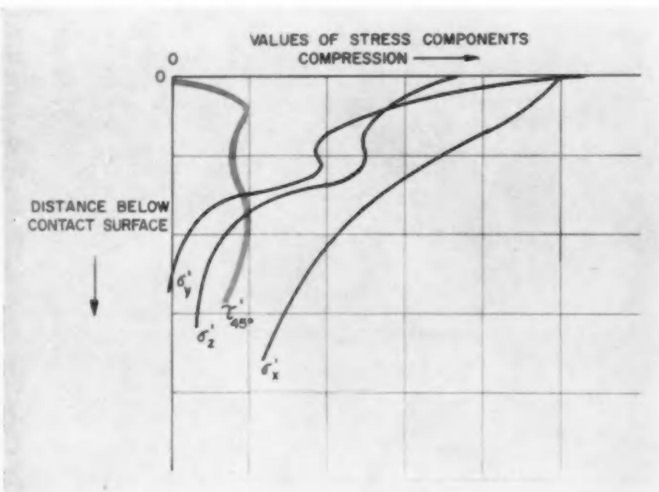


Fig. 4 — Typical stresses due to normal load and residual stresses.

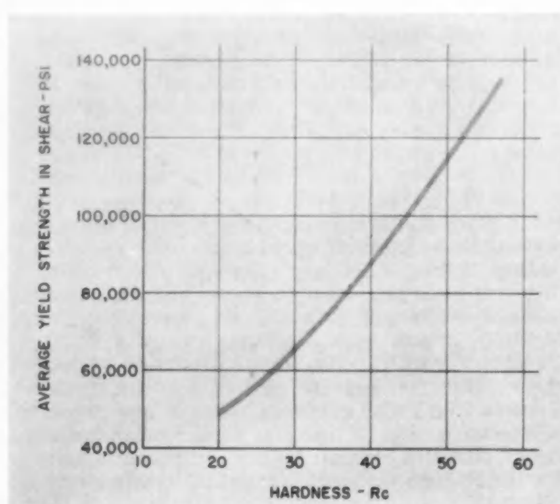


Fig. 5 — Curve for converting hardness to average yield strength in shear.

Carburized and Hardened Gears

. . . continued

the added assumption that tensile yield strength is proportional to yield strength in shear, a curve is drawn for converting hardness to average shear yield strength as shown in Fig. 5. The hardness gradient curve can then be converted to a shear yield strength curve, shown in Fig. 6.

Calculating stress/strength ratio

When a shear stress curve (τ^1_{45}) as in Fig. 4, and a shear strength curve (Fig. 6) are plotted on the same coordinate axes, the result is similar to Fig. 7. The individual shapes of these curves as well as their relationship to each other are typical for carburized and hardened gears. The shear stress is shown to be very near a maximum value at a depth where the hardened case meets the softer core material. In the same region the shear stress is greater in proportion to the shear strength than at any other depth. This is shown in Fig. 8.

At about the depth where case and core meet, the stress/strength ratio reaches a maximum value, but compressive residual stresses are small enough to be negligible. The maximum value of the stress/strength ratio for a particular gear can be obtained by simply computing the shear strength and τ_{45} due to load, both where case and core meet.

Example of application

The foregoing analysis was applied to a number of test gears. They included experimental final-drive and bevel gears tested in tractors and transmission gears tested in 4-square machines. All were made from SAE 8600 or 8700 steel and were

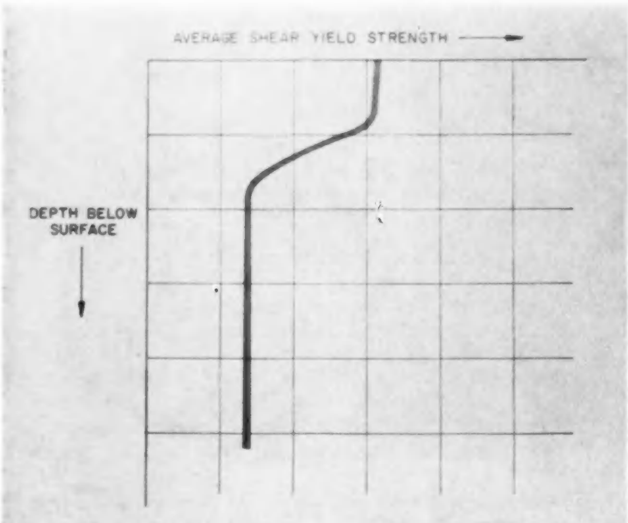


Fig. 6—Typical shear yield strength curve for a carburized and hardened gear.

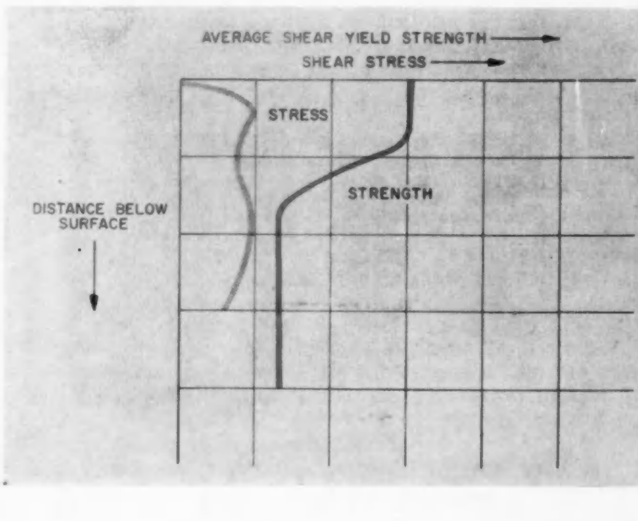


Fig. 7—Subsurface shear stress compared to shear yield strength of a carburized and hardened gear.

Carburized and Hardened Gears

. . . continued

carburized and hardened. Diametral pitches ranged from 2 to 3. Some had failed by case crushing, some by pitting, and some showed no sign of failure.

The maximum stress/strength ratios, varying from 0.30 to 0.70, are plotted in Fig. 9, arranged in descending order. The results fall into three categories or zones which are only approximate and can be defined more exactly on a statistical basis when more tests results are available.

It is felt that maximum stress/strength ratio should be no more than 0.55 in order to prevent case crushing. This critical value applies only to case crushing and not to pitting fatigue which, as Fig. 9 reveals, can occur below the value of 0.55.

If the analysis indicates the probability of case crushing, one of the remedies is to increase the strength of the material in the case-core boundary which can be done by increasing case depth or core hardness, or both.

It appears, therefore, that when resistance to case crushing is considered, case depth is not necessarily related to tooth thickness, but it more strongly influenced by the radii of curvature at the point of contact, the load, and the core hardness of the material.

To Order Paper No. 220B . . .
from which material for this article was drawn, see p. 6.

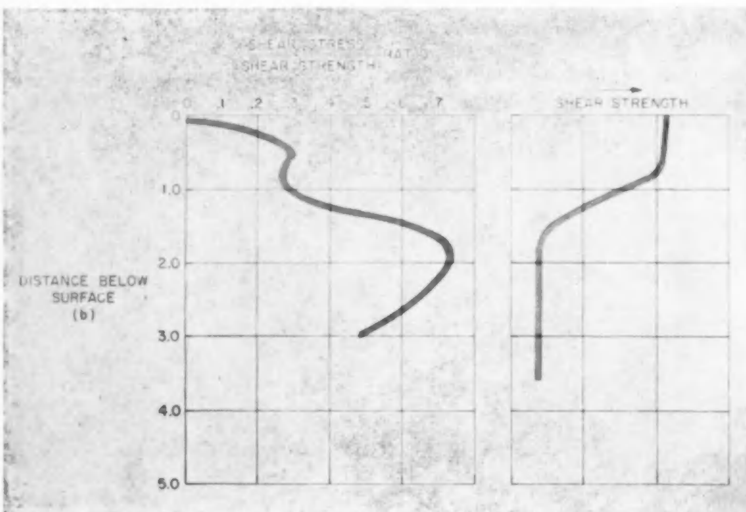


Fig. 8 — Ratio of shear stress to shear strength plotted against depth below the surface.

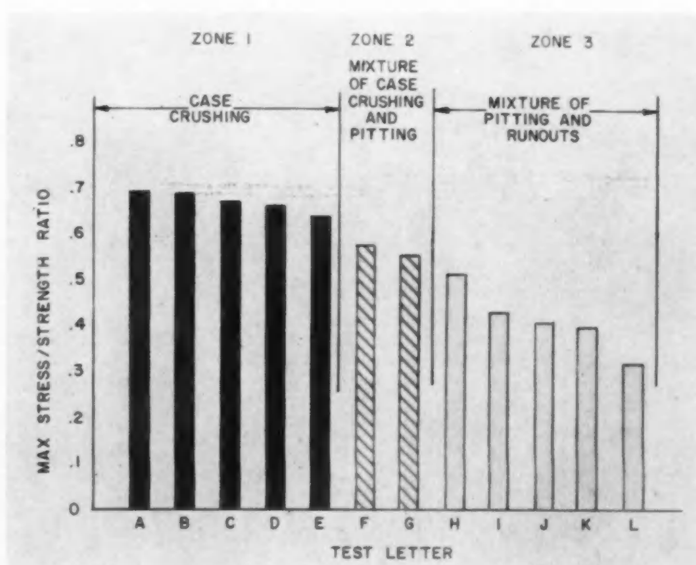


Fig. 9 — Experimental gears plotted according to maximum stress/strength ratios show 0.55 to be critical value above which case crushing occurs.

3-Speed Tandem Drive Axles

promise truckers faster shifting and sustained higher road speeds

Based on paper by

R. K. Nelson

and

L. J. Valentine

Eaton Mfg. Co.

A 3-SPEED tandem axle coupled with a 5-speed transmission goes the 2-speed tandem axle one better by providing an intermediate range. And with its short steps it decreases shifting time and makes possible operating at optimum road speeds. Such axles are on test in trucks and their satisfactory performance promises the advantages of short wheelbase coupling, operation at consistent high horsepower, and easy operation for the driver.

The possibility of a 3-speed tandem axle became apparent while developing the power divider interaxle differential for 2-speed tandems. The power divider unit distributes power equally from the transmission to the rear axles, both forward and rear. The interaxle differential unit (mounted on the input shaft) functions like a conventional axle differential to provide differential action between forward and rear rear-axles. The differential drive in each drive axle provides differential action for its respective wheels.

Shift mechanism

The system which permits shifting both axles to low or to high, or one axle to high and the other to low to give the intermediate range, consists of two solenoid valves controlled by a 3-position switch mounted on the gearshift lever.

When the driver moves the control switch to high range positions, current to the solenoid valves allows air to pass through the valves to the air shift diaphragm on both tandem drive axles. This puts the axle in readiness to shift to high range when torque on gears is relieved by closing and opening the throttle, or declutching. Moving the switch to low allows both solenoids to shut off the air supply and air pressure in the shift units bleeds back through the solenoids and pressure to the units is released. Shift to low is effected when the torque of the gears is relieved in similar manner as in shifting to high.

For shifting to intermediate range, one solenoid valve shuts off air to the rear rear-axle, shift unit while the other solenoid allows the forward axle shift unit to retain its air. Then, when closing and opening the throttle or declutching relieves torque on the gears, the rear rear-axle is shifted to low while the forward rear-axle remains in high.

Gear ratios

The difference between high and low range in a 2-speed tandem axle is about 36%. Therefore, an intermediate range in a 3-speed tandem would represent a reduction of about 18%. For example, a 3-speed tandem axle with a 4.56/1 ratio in high and 6.21/1 ratio in low would have an intermediate range ratio of 5.38/1. Shifting the axle to all three ranges in each transmission gear is not essential. In some instances it may be necessary to skip one or two ranges in the transmission gear. This can be done by simply moving the control switch to the desired axle range and completing the shift as descirbed.

To Order Paper No. 250A . . .
from which material for this article was drawn, see p. 6.

Effect of Traction on Cornering Force

Method has been devised to measure the relationship quantitatively

Based on paper by

Charles A. Freeman, Jr.
Ford Motor Co.

THE RELATIONSHIP of the effect of traction on cornering force which has proved illusive in the past has now been measured quantitatively by a relatively simple method. The method verifies a theoretical analysis for predicting the effects, made earlier and separately by Walter Bergman of the Ford Motor Co.

How the method works

The method uses a passenger car in straightline motion with an externally applied lateral load. A test tire is mounted on a rear wheel of the car to receive driving torque from the engine. A field dynamometer towed at various angles provides the required load for development of both traction force and cornering force.

The test car is driven straight ahead on a flat concrete highway with the driver maintaining constant engine speed. The dynamometer is driven at constant road speed. Holding constant engine and road speed gives constant driving torque on the test wheel, so that traction force becomes the controlled variable. The opposite wheel is uncoupled from the drive-train so that all power can be applied to the test wheel to get high torques.

At the start of a run, the dynamometer is directly behind the test car. As the vehicles move forward, the dynamometer swings slowly into the adjacent lanes while the dynamometer operator increases the brake load to maintain constant speed. The component of the towline tension in the direction of motion of the test vehicle is effectively constant so traction force is effectively constant. The lateral component of the towline tension increases as the dynamometer swings into adjacent lanes so a full range of cornering forces and slip angles is generated. A number of runs at appropriate increments of traction force provides the complete set of data relating traction force, cornering force, and tire slip angle.

Test with free rolling tire

To generate cornering forces on a free rolling tire, the driveshaft is removed from the test vehicle, and a third vehicle is added to the procession, attached by cable to the test vehicle to tow it and the dynamometer. A similar procedure in all respects then follows. The traction in this case is really a drag force and is negative — the force required to rotate the tire and the rear axle mechanism.

A merit of this procedure is the generation of a full range of cornering forces and slip angles for a given value of traction force in a single run; yet, the data may be regarded as steady-state data because of the low rate of change of the applied cornering force. Only a low level of lateral weight transfer occurs because the tow cable is attached to the axle housing 4 in. above the pavement.

Another advantage is the short time required to get complete test data — 18 runs in a single day. Tire temperature rise from start to finish of the highest power run was less than 10 F and total tread wear for all runs was less than 0.10 in.

Measurements and calculations

Of the three quantities of prime interest — traction force, cornering force, and tire slip angle, only the last is measured directly. Traction force and cornering force are calculated from effective tractive effort, lateral tire force, and tire slip angle as shown in Fig. 1.

Lateral tire force is measured directly, but effective tractive effort is the summation of towline tension, angle of the towline to the rear axle, and the frame to axle forces at the lateral locations of the right and left springs. The summing of these forces is accomplished by Fig. 2 — a diagram of the hor-

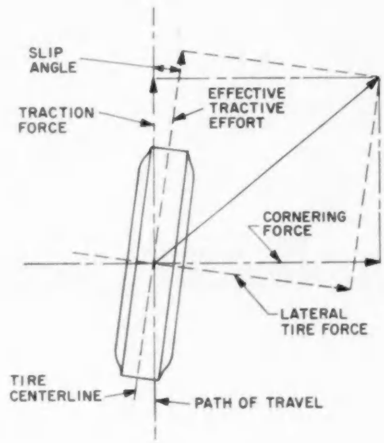


Fig. 1 — Diagram of horizontal road to tire forces. Traction and cornering forces are calculated from effective tractive effort, lateral tire force, and tire slip angle, as shown here.

TRACTION FORCE = EFFECTIVE TRACTIVE EFFORT x cos SLIP ANGLE – LATERAL TIRE FORCE x sin SLIP ANGLE

CORNERING FORCE = LATERAL TIRE FORCE x cos SLIP ANGLE + EFFECTIVE TRACTIVE EFFORT x sin SLIP ANGLE

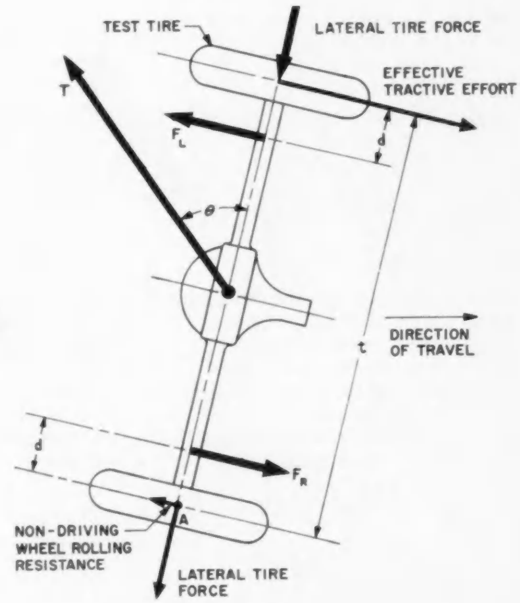


Fig. 2 — Horizontal forces on rear-axle assembly during test. The vectors identified by heavy arrows are measured quantities.

EFFECTIVE TRACTIVE EFFORT =

$$T\frac{\sin\theta}{2} - F_R\frac{d}{t} + F_L\left(1-\frac{d}{t}\right)$$

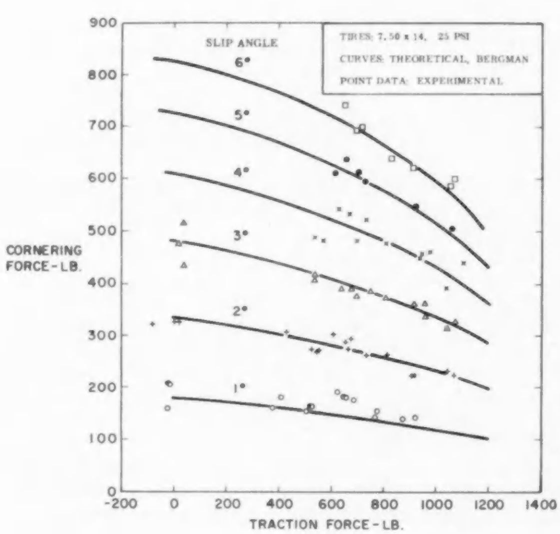


Fig. 3 — Cornering force versus traction force at various slip angles, showing close correlation between curves (theoretical) and point data experimental.

Effect of Traction on Cornering Force

. . . continued

izontal forces on the rear axle assembly. When we sum the moments about point A and operate on the resulting equation, we obtain an expression of effective tractive effort. This expression, presented in the figure, permits calculation of effective tractive effort from measured quantities.

Seven measurements are taken during the test, as follows:

- Tire slip angle.
- Lateral tire force.
- Axle to frame force, right side.
- Axle to frame force, left side.
- Towline tension.
- Towline angle.
- Car speed.

The last three quantities are measured with conventional equipment — a resistance strain gage drawbar scale, a circular slide wide potentiometer, and a fifth wheel. The first four quantities require special equipment for measurement.

Test results

No high-gain amplifiers are used. Each instrument is designed to have a d-c voltage output adequate to operate a galvanometer at a reasonable and usable sensitivity. Test data are reduced from the oscillogram at 1-deg slip angle increments. The accuracy of measurement of each function relies more on consistent calibration than on instrument sensitivity. Calibrations are performed before and after each set of runs to check instrument stability and limit instrumentation errors to a maximum of $\pm 1\%$.

The calculation of effective tractive effort (Fig. 2) compounds per cent instrument error because multiplication of two measured quantities is involved — towline tension and towline to axle angle. Thus error can run as high as $\pm 1\frac{1}{2}\%$. The calculations illustrated in Fig. 1 do not introduce appreciable error because adjustment of forces for slip angle is relatively small and slip angle is measured to a high degree of accuracy.

The final plot of cornering force versus traction force is obtained after applying the equations in Fig. 1 on the lateral tire force and effective tractive effort data.

Theoretical versus actual

A graph of cornering force versus traction force is shown in Fig. 3, together with the theoretically predicted curves. Each point of the test data plotted is a single observation, not the average of many runs. The scatter, which amounts to $\pm 5\%$ of the maximum values of lateral tire force, is due mainly to test anomalies, such as winds, road surface variations, and slight vehicle accelerations.

To Order Paper No. 186B . . .
from which material for this article was drawn, see p. 6.

Paradoxical products—

Based on paper by **Earl L. Will** Monsanto Chemical Co.

GEAR LUBRICANTS are paradoxical products. The performance requirements and limitations imposed upon them are strongly contradictory. They must react rapidly with metals, but not corrode them. They must be stable at temperatures to 300 F, but reactive at higher temperatures. They must form high strength films for carrying loads, but low strength films for reducing friction. They must be high in performance, but low in cost.

Gear lubricants are paradoxial products in another sense, in that the petroleum base oil which in popular language has become almost synonymous with "lubricant" is of secondary importance in the functionality of the products. In many lubricant applications the characteristics of the petroleum hydrocarbon base are of primary importance. Additives are used to reinforce properties of the original lubricant, and to a lesser extent, to impart new and desirable properties not originally present to any significant extent. In many gear lubrication services the pressures prevailing eliminate a fluid film, and lubrication depends on the additives rather than the oil.

This article discusses the role that additives play in making difficult lubrication problems solvable, and the guides that are available for selecting gear lubricants.

Regular-Type Gear Lubricants usually contain no additives, with the possible exception of a small amount of foam inhibitor. They are comprised of straight mineral oils selected on the basis of viscosity, channeling characteristics, stability and oxidation resistance, and lack of foaming characteristics.

Worm-Type Gear Lubricants are designed to protect worm gear sets in which the tooth pressures and rubbing velocities are such that they cannot safely be lubricated by regular-type gear lubricants. Additives are used to enhance a poorly defined property of the fluid commonly referred to as "oiliness, lubricity, or film strength." The oiliness additives are oil soluble compounds containing highly polar functional groups which will form strong bonds to gear metal surface and "tails" which extend into a film. Slippage can occur between these molecular tails. Chemically they may be oxidized oils, animal or vegetable fats, sulfurized fats, aliphatic amines, esters. All of these materials will form films on metal surfaces which adhere much more firmly than

Lubricants for Gears

those from the base lubricant. Choice of the additive is dictated by effects on stability, oxidation, viscosity, cost, and the desirability of incorporating other functions, such as rust inhibition.

Mild-Type Extreme Pressure (E-P) Gear Lubricants are designed to have load carrying properties suitable for many automotive transmissions and spiral bevel differentials under severe conditions of speed and load. It is in designing this type of lubricant that the additive chemist and lubrication engineer run head on into grossly contradictory requirements. Since the same problems, except in exaggerated form, also arise with the **Multipurpose-Type Gear Lubricant (API Service GL-4)** the additive requirements of both can be treated simultaneously.

Extensive analyses of field experience and dynamometer testing on full-scale equipment have shown that two different boundary lubrication phenomena are involved in the operation of hypoid gears. Either one may predominate, depending on design and operating conditions, but the probabilities are high that both will be operative during the life of most gear units. A really satisfactory lubricant must therefore provide protection against the scoring-type damage associated with high sliding velocities and excessive wear that accompanies extremely heavy loading at more moderate velocities.

Under conditions of high sliding velocity between gear faces, temperatures over 1000 F have been measured at the metal surfaces. At these temperatures welding and transfer of metal between surfaces occurs as the welds are broken. The result is a scored gear. Certain e-p additives react chemically with the gear metal surfaces to form an easily sheared inorganic film which acts as the boundary lubricant. Ideally this film should not form at oil sump temperatures, but only at the elevated temperatures that exist under boundary lubrication conditions. An overly active additive may chemically remove metal from a gear, resulting in gear failure. At the critical temperature, however, the reaction must be fast enough to replace the film between revolutions of the mating gears.

Among the classes of organic compounds which have been reported to be helpful in combating scoring under high speed conditions are:

- Chloro organics
- Active sulfur in noncarboxylic compounds
- Relatively inactive sulfur in noncarboxylic compounds
- Relatively inactive sulfur in carboxylic acids
- Phosphite acid esters
- Thiophosphate neutral esters

Under high load conditions, excessive wear occurs rather than scoring, and indicates predominance of a different phenomenon. In the absence of the hydrodynamic film, metal-to-metal contact occurs between surface asperities. When pressures exceed the tensile strength of the metal, minute particles are broken off. The process continues and an early failure of the unit ensues. The action of certain additives may be explained by their known ability to form low melting point alloys or soaps with surface materials. These alloys flow into irregularities in the sliding surfaces, resulting in a broader load distribution and decrease in surface temperatures.

Among the classes of compounds which are definitely effective under high torque conditions are:

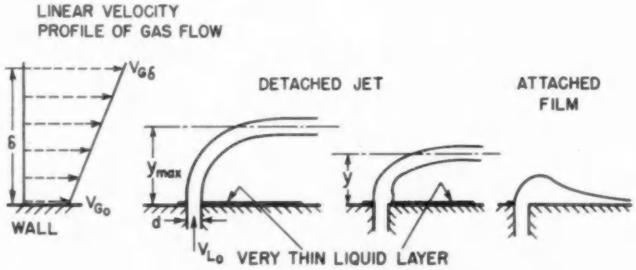
- Carboxylic acids and esters
- Carboxylic acids and esters containing relatively inactive sulfur
- Phosphite acid esters
- Thiophosphate neutral esters

The lists for high-speed and high-torque operations are very similar, in that it may seem readily possible to select materials which are effective in both services. Remember, however, that the lists are for classes of compounds. When specific chemicals are tried, it will be found that some that are helpful in high-speed conditions will be harmful or inactive under heavy-load conditions, and others that are helpful under heavy-torque conditions will be harmful or inactive under high-speed conditions.

Materials that are compatible with both service requirements may be eliminated from commercial use when the other properties that a heavy-duty gear lubricant must have are considered. Stability, corrosion protection, low-temperature fluidity, foaming, cost, toxicity, odor, compatibility with other lubricants, and compatibility with many petroleum base stocks and reaction with elastomeric seals are some of the factors that must be weighed in addition to the ability of the additives to give gear tooth surface protection under both extreme load and speed conditions.

To Order Paper No. 217B . . .
from which material for this article was drawn, see p. 6.

Fig. 1
Attachment process of liquid film in three successive steps for decreasing liquid flow rate at otherwise constant conditions.



Film vaporization combustion is applicable to gas turbine burner

Principle is based on M-system already successful on diesels. New system is expected to minimize deposits and smoke. Specific problems involved in designing a research prototype combustor are discussed here.

Based on paper by

A. W. Hussmann, Pennsylvania State University
and G. W. Maybach, Maybach-Motorenbau

THE DESIGN of a research prototype combustor for gas turbines, based on the film vaporization principle already so successfully applied to diesel engines (see Box on pp. 46–47) meant the investigation of the following problem areas:

- Attachment of liquid films onto the inside wall surfaces of circular ducts in the presence of high-velocity gas streams flowing through the ducts.
- Spread of such films.
- Stability behavior of such films.
- Problem of simultaneous heat transfer and mass transfer between a high-temperature, high-velocity gas stream and the liquid fuel.
- Recirculation flow of hot combustion products from the reaction zone upstream into the vaporization section, and the ensuing problem of flame stability.

Film attachment

Initial investigations of the possibilities of attaching liquid films onto the inside wall surfaces of a duct in the presence of high-velocity gas streams flowing through the duct were conducted in a model. This model permitted the insertion of various injection devices, such as slots, screens, and single-hole nozzles, inclined at different angles toward the duct wall surface. The duct cross-section in the test section of the model was a flat square. The airflow in the test section was fully developed turbulent.

These initial experiments, in conjunction with conclusions established by other researchers in the field of film attachment, proved that the simplest and most effective method of attaching liquid films is to introduce the liquid through holes drilled through the duct wall perpendicularly to the surface. Then the film becomes attached by the action of the high-velocity gas stream.

Fig. 1 shows, in three successive pictures (from left to right), the attachment process of a liquid jet entering perpendicularly into a gas stream of linear velocity profile when, at otherwise constant conditions, the liquid flow rate through the injection hole is reduced.

The liquid jet enters the gas stream perpendicu-

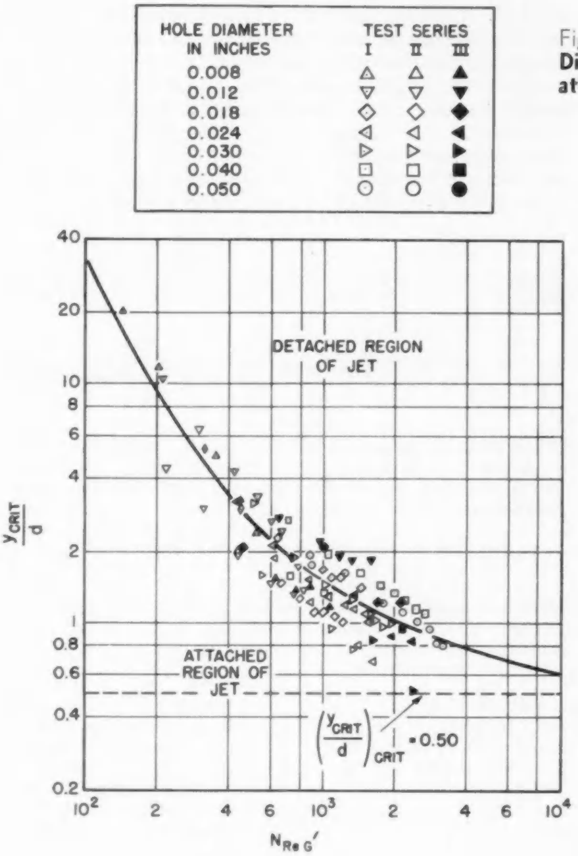


Fig. 2
Dimensionless plot of film attachment criterion.

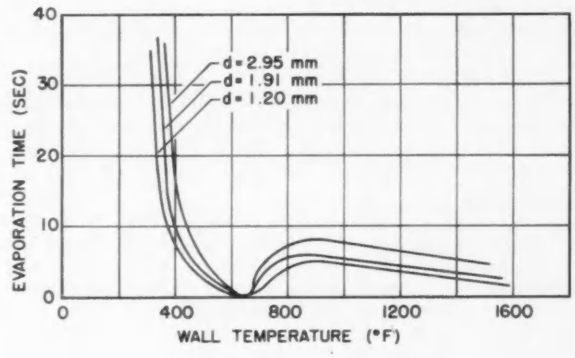


Fig. 3
Evaporation time of lens-shaped drop of kerosene on a heated surface (tests by Tamura and Tanasawa).

larly, becomes deflected in the direction of the gas stream by the action of the radially increasing gas velocity, and reaches asymptotically the maximum penetration distance y_{max}. For a reduced liquid flow rate, at otherwise constant conditions, the maximum penetration distance decreases and the base of the jet in the immediate vicinity of the wall increases in diameter. With a further minute reduction of the liquid flow rate, the action of the gas flow causes the thickened base of the jet to be swept downstream, thus attaching the jet to the wall surface.

A criterion to express the conditions at which film attachment can be obtained was found to consist of a relationship between the maximum penetration distance of the liquid jet (that is, the potential maximum penetration distance) and the gas stream Reynolds number based on the jet diameter (that is the injection hole diameter) and the gas flow velocity at the wall. The criterion was found to be independent of surface tension forces or interfacial tension forces between liquid, gas, and wall surface, independent of gravitational forces, and independent of the duct geometry.

The experimentally established criterion for film attachment is expressed in Fig. 2. The ordinate is chosen to be the dimensionless critical maximum penetration distance for film attachment, and the abscissa is the Reynolds number of the gas stream (based on the injection hole diameter and gas velocity at the wall). The critical penetration distance is the maximum distance at which incipient film attachment can still be obtained.

The curve in Fig. 2, drawn through the average of several plotted test data, reveals clearly the trend of the film attachment mechanism. The test data were obtained for several different injection hole diameters, various liquids, various liquid supply rates, and various gas stream velocities and geometries. The critical maximum penetration distance for a liquid jet to become attached to the wall surface approaches a minimum value of $y_{crit}/d = 0.50$ for increasing gas Reynolds numbers. For this value the jet must necessarily touch the wall surface right at the orifice; that is, the jet remains attached. If the maximum penetration distance of the jet exceeds one-half of the injection hole diameter, the conditions at which,

for smaller gas Reynolds numbers, film attachment can still be expected can be obtained from Fig. 2.

If a liquid film is to be attached from radial injection holes to the wall surface of a circular duct, and if the flow pattern of the high-velocity air is a swirl, a liquid film can still be produced, although the liquid jet may be detached. In this case, the film spreading mechanism is based on the centrifugal force acting on the liquid, which is forced to flow in a curved path. In a straight airflow, the liquid jet propagates in a plane that contains the center axis of the duct; in a swirling airflow, the liquid jet becomes deflected from that plane and is impinged back to the wall surface of the duct.

Film spread

A coherent and homogeneous spread of the liquid film is of importance to make economical use of the available vaporization surface.

The experiments revealed that the spreading of a liquid film from holes positioned perpendicularly to the surface onto which the film is to be spread depends on two factors: the surface tension coefficient of the liquid, and the method of film attachment.

The effect of the surface tension forces is such that, if the difference of the surface tensions of the solid and the liquid is equal to or larger than the interfacial tension between the solid and the liquid, the contact angle vanishes and the liquid spreads nearly infinitely as film. Jet fuel and most other petroleum oils satisfy this condition for film spread for most clean metallic surfaces. However, the surface tension coefficient of water, for instance, combined with interfacial tension between the solid and the water, is so much larger that the contact angle normally assumes a finite value and no film is spread, unless mechanical means are applied.

The other factor influencing film spread is the method of attachment, that is, whether the film is attached by a straight gas (air) flow or a swirling gas flow. The experiments proved that attachment by means of a swirling airflow has a generally superior effect on the film spread of the liquid of high surface tension. However, the liquid injection holes can be designed so that, within a certain range of operation, the liquid film is spread as homogeneously by a straight airflow as by a swirling airflow. In order to make use of this effect for liquids of high surface tension coefficient, the injection holes have to be placed so closely that the liquid streaks leaving the injection holes contact each other, thus forming a continuously spread film surface.

Film stability

Several investigators have observed that, when liquid films are exposed to high-velocity gas flows, surface disturbances of a wavelength of approximately 10 times the film thickness occur for all liquid flow rates. The scale of these disturbances was found to decrease with increasing gas Reynolds number. Relatively smooth and stable film surfaces are obtained only when the film thickness, and consequently the liquid flow rate, does not exceed a certain critical value. If the thickness is greater than this critical value, severe dis-

New combustor

THE film vaporization combustor for gas turbines discussed here is based on the M-combustion-system invented by Dr. J. S. Meurer for diesel engines (*SAE Journal*, September, 1955, p. 18)

In contrast to conventional diesel combustion of atomized fuel, the fuel is not atomized in the M-system, but is spread as a film on the relatively cool wall of the combustion chamber. A vigorous air swirl with a peripheral velocity in the direction of the fuel jet helps to centrifuge the fuel against the wall and so spread it as a film, as shown in Fig. A.

The fuel issuing from a high-pressure injector does not, however, reach the wall in an entirely solid jet. A small part is unavoidably atomized. The larger droplets are centrifuged against the wall in a short trajectory, while the smaller droplets travel a longer distance with the air. During that time they evaporate partly and, after the ignition delay period, ignite. Since most of the fuel is on the wall, the resulting initial pressure rise is small. The burning droplets are carried around by the air, igniting the vapor-air mixture forming off the wall before the self-ignition temperature is reached and before the fuel can crack. The total reaction is now accelerating without getting out of control, with more and more heat available for evaporation. In the strong centrifugal field of the swirl, the lighter combustion products move inward, while the fresh air moves outward to sustain combustion in the vapors forming off the wall. Recent investigations using high-speed pictures show very clearly the strong inward radial velocity component produced by combustion in a swirl.

In transposing the M-system into a gas turbine combustor the following elements were retained:

1. The fuel is spread as a film on the wall of a "vaporizer tube."

2. Primary air is flowing through this tube with a vigorous air swirl.

3. The walls of the tube are kept relatively cool and the heat of evaporation is taken largely from the combustion itself by means of recirculating combustion products in the core of the air swirl.

Fig. B shows schematically such an arrangement. A portion of the air from the compressor flows as a secondary stream around the vaporizer tube and mixes later in the flame tube with the combustion products. Another part, the primary air, flows through swirl-producing vanes and the vaporizer tube into the flame tube. The fuel is intro-

turbances develop over the entire film surface, with the effect that some of the liquid becomes directly entrained by the gas flow in the form of droplets. The incipient point for film disturbance is independent of the gas flow Reynolds number or the mass rate of flow of the gas.

For liquid films in the presence of straight, concurrent gas flows, at least two stability criteria exist. An entirely different behavior of the stability of liquid films, however, was observed in the experiments when the films were applied with the aid of swirling high-velocity gas flows.

At swirling airflows and liquid flow rates corresponding to thin films, the film surfaces did not behave differently from corresponding experiments with straight airflows. For increased liquid flow rates, however, although the film surface was covered with severe disturbances, the liquid drop-

based on M-system for diesels

duced under low pressure through a multitude of radial holes into the vaporizer tube and is spread onto the inner surface of that tube by means of the air swirl. The wall separating the primary-air duct from the secondary-air duct is cooled by secondary air on the outside and shielded against conductive heat transfer from the adjoining reaction zone.

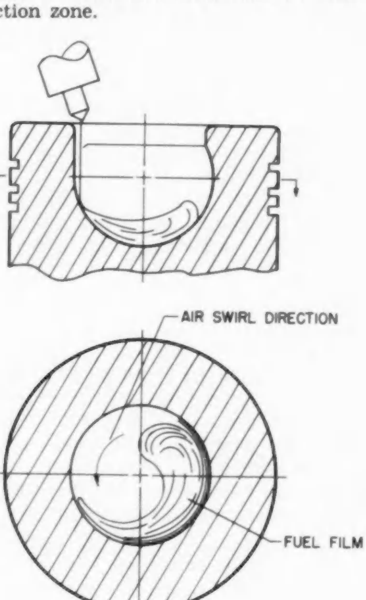


Fig. A
M-system has been successfully applied to diesel engines.

At the end of the vaporizer tube the air swirl expands into the reaction zone, where the flame stabilizes, and further into the flame tube. The vortex-type flow of the primary air produces a low-pressure region in the core of the vortex, leading to a strong recirculating flow of hot combustion products back into the vaporizer tube. While being recirculated the hot combustion products mix with the primary air and increase its temperature. The heat necessary for the vaporization of the liquid fuel is transferred from the primary-air swirl to the liquid film by turbulent convection. Immediately upon its formation, the fuel vapor diffuses into the primary-air swirl by convective mixing, thus ensuring that the fuel vapor-air mixture leaving the vaporization zone and entering the reaction section is a thoroughly premixed combustible mixture.

The inherent advantage of prevaporizing the fuel from a film rather than from droplets lies in the fact that a comparatively large relative velocity between the fuel and the primary air can be maintained when the fuel is spread as film. If the fuel is atomized, an initial relative velocity between the fuel droplets and the air will very quickly diminish to a negligible value because of the large drag of the droplets, and the droplets will float in the air with a velocity approximately equal to the air velocity. A large relative velocity between the fuel and the air increases the heat-transfer and mass-transfer coefficients at the liquid-gaseous interface, and increases the rate of mixing of the fuel vapors and the air, thereby reducing the length required for the vaporization section and for the entire combustor.

A further and most important advantage of film vaporization is the fact that the fuel and fuel vapors can be kept below the cracking temperature during the mixing process.

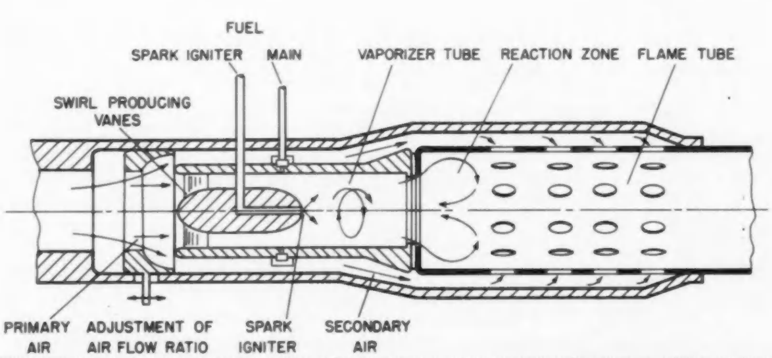


Fig. B
Schematic diagram of film vaporization combustor for gas turbines.

lets that eventually separated from the film surface were immediately centrifuged back to the film surface. This centrifuging effect on the droplets is due to the motion with a peripheral component which the swirling airflow imparts to the droplets as they separate from the film surface.

The swirling gas flows produce films that are stable over the entire range of film thicknesses, irrespective of the intensity of disturbances on the film surface. If, however, the films are applied by means of straight airflows, a film stability criterion must be observed in order to avoid instabilities.

Film vaporization

The heat for evaporation of the liquid fuel film is taken from the vaporized tube wall as well as from the air mixed recirculated combustion products.

The influence of the wall temperature on the evaporation rate of a fuel film has not been adequately studied. Tamura and Tanasawa have investigated the evaporation of a droplet on a heated quartz plate. Fig. 3 shows as an example their results with kerosene. Plotted are the times for complete evaporation against the wall temperature. As expected, the time decreases at first with increasing temperature until it reaches a minimum. If the wall temperature is further increased, the time needed for complete evaporation rises again to a time many times greater than that which was needed at the lower temperature. The reason is the well-known Leidenfrost phenomenon, insulating the liquid drop from the hot surface by a vapor cloud. It is most likely that, for film vaporization in a strong swirling flow, similar curves would be obtained but that the critical temperatures for highest vaporization rates would be considerably higher. The insulating effect of the Leidenfrost phenomenon would partly be balanced by a strong centrifugal field.

These tests show that it would be advantageous to keep the wall temperatures above the vaporization temperature of the main fuel fraction. Higher temperatures not only slow down the rate of evaporation but may lead to partial cracking of overheated fuel vapors and consequently slowed down reaction rates for the later combustion.

In addition to the heat input from the wall, the heat and mass transfer between the fuel film and the air govern the evaporation rates. Several theoretical and semi-empirical methods exist for calculating vaporization rates from liquid films spread on the inside wall surfaces of ducts when the high-temperature and high-velocity airflows inside the ducts are fully developed turbulent and straight flows. As yet, no effective theory has been developed that would permit the prediction of film vaporization rates for the case of gas flows different from fully developed straight turbulent flows, such as swirling gas flows. Since the swirling gas flow is of special interest to the film vaporization combustion principle, a semi-empirical method was, therefore, developed to account for the effects of the swirling gas flow on the film vaporization process.

Kinney et al. proved experimentally that the well-known dimensionless equation for heat transfer under conditions of forced convection (see Appendix for nomenclature):

$$N_{Nu} = a(N_{ReG})^m (N_{PrG})^n \tag{1}$$

is applicable also to superimposed heat and mass transfer between a fully developed turbulent high-temperature air-flow and a stationary, stable liquid film. The exponents were found to be $m = 1$, $n = 0$; and the proportionality factor a was found to be constant over a large range of Reynolds numbers, so that Eq. 1, with N_{PrG} = constant reduced to:

$$N_{Nu} = b(N_{ReG})$$

or:

$$\frac{\alpha D}{k_G} = b N_{ReG} \tag{2}$$

If one equates the enthalpy increase Δh of the evaporated fuel from the state of injection through evaporation to the heat transfer:

$$\dot{m}_f \Delta h = \alpha \, \Delta \, T \, \pi \, D \, L \tag{3}$$

and introduces α from the above general relation between the Nusselt and Reynolds number, one gets:

$$\frac{m_f}{\pi D} = b \cdot \frac{L \, K_G (T_G - T_s)}{D \, \Delta h} \cdot N_{ReG} \tag{4}$$

$$y = b \cdot x$$

A generalized plot of the vaporization rate per unit periphery according to the above equation showed a

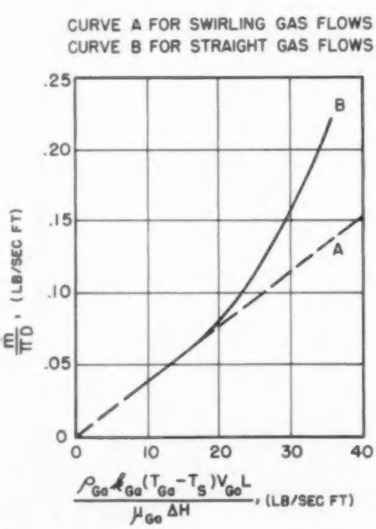


Fig. 4
Generalized plot for film vaporization rates.

straightline relation for smaller flow rates. For larger liquid flow rates (thick liquid films) the plot departed from the straightline, which was explained by Kinney et al. to be due to the increasing film instability for increased flow rates causing liquid entrainment by the airflow.

The semi-empirical method of Kinney et al. was found to be very suitable to an extension to swirling gas flows inside circular ducts. For the extension the assumption was retained that heat transfer is the process controlling mass transfer from the liquid film. In order to account for the swirling gas motion, however, the Nusselt number and the Reynolds number of Eq. 4 have to be modified.

The characteristic velocity entering the Reynolds number is generally represented by the average bulk velocity or the undisturbed velocity. For swirling gas flows inside a circular duct, however, the definition of an average axial velocity has little meaning except for determination of the axial through-flow. Therefore, the absolute gas velocity near the wall surface V_{Go} was chosen to represent the characteristic velocity for the evaporation rate in swirling gas flows inside circular ducts.

Eq. 4 can now be written:

$$\frac{\dot{m}_f}{\pi D} = 3.8 \times 10^{-3} \frac{\rho_G k_G (T_G - T_s) V_{Go} L}{\mu_G \Delta h} \tag{5}$$

The constant in the equation was determined by introducing the velocity near the wall instead of the bulk velocity. The ratio of these velocities was experimentally found by mapping the velocity profiles for the 1.75-in. diameter vaporization tube and for Reynolds numbers between 4×10^4 and 16×10^4. Fig. 4 shows a plot of Eq. 5 in curve A. For higher vaporization rates (liquid flow rates) the curve B rather than the straightline extension would hold for straight flow, indicating the influence of the film instability and entrainment of liquid. Since the film for swirling gas flows is stable up to much higher gas and liquid flow rates, the dashed extension of the straight line relation (curve A) may be considered as a conservative estimate for the vaporization rate under swirling flow. Actually, film disturbances, entrainment, and centrifuging back to the wall of entrained particles will increase the vaporization rates beyond that indicated by curve A.

Recirculation of combustion products

There are two principal types of recirculation that are used in combustion chambers to transport combustion products behind the flame front: the macroscopic or large-scale recirculation and the microscopic or small-scale recirculation. The objective of recirculating a part of the combustion products is to render stable flames which, in a given space, burn all of the fuel admitted to the combustion chamber.

Large-scale recirculation is a reverse flow induced by pressure differences as a secondary flow. Unlike microscopic recirculation, which is essentially a local region of turbulence induced by flow separation, large-scale recirculation is a reverse flow that can be utilized to transport hot combustion products over a relatively large distance behind the flame front, into the region of the primary airflow.

Various experimental arrangements of the vaporization and reaction sections of the film vaporization combustor were investigated in isothermal flow tests and in actual combustion tests for their ability to produce a sufficient amount of large- and small-scale recirculation at minimum pressure losses. In order to determine quantitatively the rates of the recirculation flows, a special probe was devised to measure velocity patterns in swirling flows inside circular ducts.

Appendix

Nomenclature

Symbol	Dimensions	Definition
α	Btu/ft²-sec-F	Heat transfer coefficient
a	—	Constant
b	—	Constant
D	ft	Duct diameter
d	ft	Liquid jet diameter
Δh	Btu/lb	Change in enthalpy from injection condition through vaporization
k_G	Btu/ft-sec-F	Average heat conductivity of bulk of gas
L	ft	Length of liquid film
m	—	Constant exponent
μ_g	lb-sec/ft²	Average gas viscosity
n	—	Constant exponent
$\dot{m}_f$	lb/sec	Weight rate of film vaporization
N_{Nu}	—	Nusselt number
N_{Pr}	—	Prandtl number
N_{Re}	—	Reynolds number
ρ_G	lb/sec²/ft⁴	Density of gas
T_G	R	Temperature of the gas near the wall
T_s	R	Temperature of the film surface
V_{Go}	fps	Gas velocity near the wall
y_{crit}	ft	Maximum penetration distance of jet for film attachment

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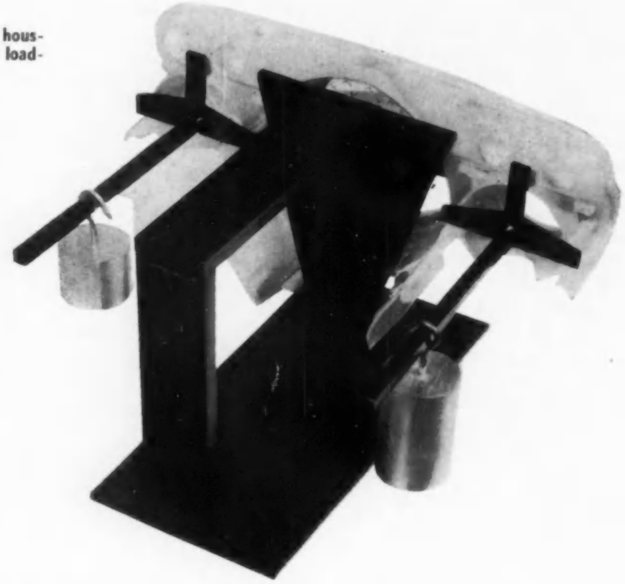
from which material for this article was drawn, see p. 6.

Fig. 1 — Prototype housing with attached loading beams.



Plastics

are replacing traditional materials in selected automotive applications

Based on report by

A. J. Carter

Chrysler Corp.

BECAUSE so many factors — temperature, chemical resistance, type of loading, molding conditions, and design — affect the properties of polymeric materials, considerable judgment beyond calculated design data is required in making the decision to use these materials for automotive mechanical applications.

This article describes the actual procedures taken in designing and testing a few unique automotive mechanical applications where plastics are the basic structural material.

Instrument housings

For some time, the plastic industry has looked longingly at the interior zinc die-cast applications in the automobile. The amount of material involved is large and any breakthrough as a replacement for this material would be worthwhile. The automotive engineers have also looked toward plastics as a possible replacement for metal castings since they offer a significant weight advantage as well as color and surface design potentialities.

With the introduction of the acetal polymer, Delrin, there was available for the first time a thermoplastic material with strength at elevated temperatures and excellent dimensional stability which are prerequisites for the instrument housing. This part, because of its location under the windshield, can reach temperatures as high as 225 F. It is in full view of the car occupants and must, therefore, maintain its original shape and fit for appearance reasons.

The cost potential of molded acetal parts also appeared very favorable when compared to magnesium and aluminum which were likewise being considered for this type application because of cost and weight advantages over zinc.

The part selected for investigation was the Valiant instrument housing, which was a zinc die casting. The design of the part requires that the switches and the integral instrument face plate and lenses with its attached instruments be mounted directly to the housing. These include, with their weights listed, the ignition switch, 7 oz; headlight switch, 6¾ oz; windshield wiper switch, 4 oz; heater control switch, 8¼ oz; the automatic transmission control switch, 2 lb–2⅜ oz; and the instrument face plate and attached instruments, 3 lb–2¼ oz. The total weight with screws and two small face plates is 7 lb–¾ oz.

Before initiating any experimental tooling, some preliminary tests were run on acetal prototype samples to determine its strength, set, and fatigue characteristics relative to the loads of the mounted instruments.

Of particular concern was the thread strength of the mounting bosses. These were found by drilling acetal blocks with holes to simulate the molded-in

holes in the bosses. Two thread types and two hole sizes were evaluated. Drill taps were inserted to a specified depth and the load necessary to strip the threads was determined. The rate of loading was 0.1 in. per min. The effect of the removal of a screw and its reinsertion into the same threaded hole was also ascertained. The following results were obtained:

Type of Thread	Hole Diameter, in.	Depth of Thread, in.	Times Inserted	Stripping Load, lb (Avg. 4 Samples)
10–24NC	0.161	⅜	1	925
			2	850
10–32NF	0.166	⅜	1	1050
		¼	1	580

A comparison between the thread strength of acetal polymer, zinc, and magnesium die-cast materials was also run. The results below indicate that all three have strengths well in excess of what is needed to hold the parts involved:

Material	Type of Thread	Hole Diameter, in.	Depth of Thread, in.	Stripping Load, lb (Avg. 3 Samples)
zinc	¼ in. – 20 NC	0.211	0.4	2900
magnesium	¼ in. – 20 NC	0.201	0.4	1600
acetal	¼ in. – 20 NC	0.212	0.4	1400

The next step was to determine the ability of the acetal polymer to adequately support loads at elevated temperatures without taking a serious permanent set. A prototype housing mounted into a holding fixture and two loading beams, which were welded to adapter plates, were attached onto each opening by means of three self tapping screws for each plate (Fig. 1). The distance from the top of the base of the holding fixture to the bottom of each beam was measured. The beams were then loaded at a distance of 5 in. from the face plate of the beam with 3 and 6 lb., respectively, and another measurement taken after 15 min. of elapsed time. The assembly was then placed in an oven at 212 F and the corresponding distances were measured periodically. After 312 hr exposure, the weights were removed and the assembly was allowed to stand for 24 hr at room temperature after which a final reading was made. As indicated below, the total sets for the 3 lb and 6 lb loads were only 0.04 in. and 0.16 in. respectively.

Time, hr	Temperature, F	Distance Between Base and Beam, in. 3 lb	6 lb
0	75	7.25	7.04
0.25	75	7.19	6.82
24	212	7.14	6.64
48	212	7.14	6.60
72	212	7.14	6.60
168	212	7.13	6.55
192	212	7.13	6.55
312	212	7.13	6.52
24 recovery	75	7.21	6.88

To obtain some information on the fatigue properties of acetal, the loading beam was again mounted to a prototype housing by means of three self tapping screws. The assembly was mounted to a Sonntag, Model SF-2, fatigue testing machine. This machine produced a constant, vertical, vibrating force by means of a motor driven, adjustable, rotating unbalanced mass.

The subject part endured a total of 12,153,000 load cycles without failure. The first 1,050,000 cycles were accumulated under a machine load of ±1.5 lb (±0.062 in. amplitude measured 5 in. from the adapter plate). The remaining 11,103,000 cycles were accumulated under a machine load of ±3.0 lb (±0.125 in. amplitude measured 5 in. from the adapter plate). The self tapping screws were torqued 15–20 in.-lb prior to testing and showed no appreciable loss in torque after the 12,153,000 cycles.

These test results were so gratifying that a mold was made to produce parts for molding data as well as further vibration studies, hot and cold temperature cycling tests, and road test evaluation. The quality of the die was such that when all tests were completed, production experience could be gained if warranted.

The design of the plastic housing was a duplication of the zinc die-cast housing except that two

Instrument housings control switches, and gears are typical new plastics applications. Proper design, material selection, and fabrication techniques provide the properties required by automotive mechanical parts.

designing with plastics

Plastics . . . continued

reinforcing webs between the two upper mounting studs and the housing outer shell were reduced from 5½ in. in length to about 1¼ in. The reduction was necessary to stabilize the male die plug in this location and to eliminate potential shrink marks. It was felt that only minor changes in design would be needed to reinforce any areas that might be weak as indicated by fatigue testing.

Vibration tests were carried out as soon as parts were available. The complete assemblies of all instruments and switches were mounted onto the housing as shown in Fig. 2. This assembly was bolted to the instrument sheet metal panel and then vibrated in an appropriate testing machine at 10–25 cps for 72 hr. This vibration cycle simulates the most severe conceivable driving condtions. No failure occurred, but it was evident that changes were necessary because of the excessive vibration set up in the instruments. The unit was then vibrated at 50–65 cps to accurately locate the weak spots by failing the part.

Fig. 2 — Rear view of instrument housing assembly.

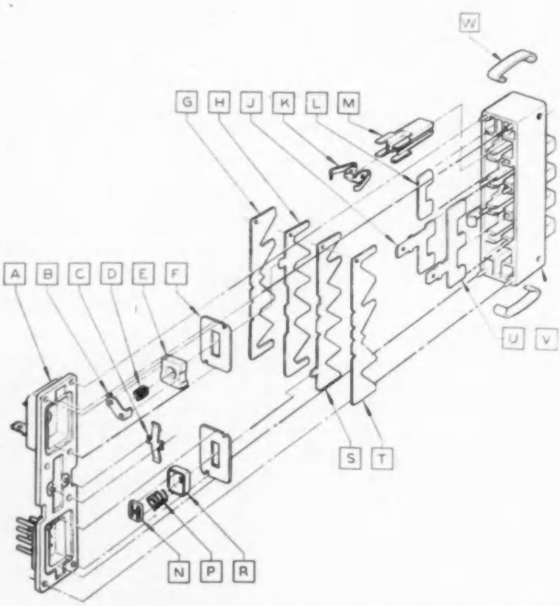


Fig. 3 — Assembly drawing of heater-air conditioning switch.

Since failures occurred in the switch mounting flanges, it was agreed to increase their wall thickness from 0.060 to 0.100 in. It was also apparent that additional webbing was needed to rigidize the upper automatic transmission control switch mounting boss and the upper heater control switch mounting boss.

The height of the bosses used to mount the housing to the instrument panel was found to be critical because distortion of the housing resulted when the bosses were too short. This factor is not critical with the zinc die-casting since it is rigid enough to force the sheet metal to conform to the die-cast shape.

Parts molded with the above corrections successfully passed the 72 hr vibration test at 10–25 cps and the instruments' vibration was considered satisfactory.

The assembly of switches, housing, and instrument panel was subjected to a hot and cold cycle treatment for three weeks to determine any mounting deficiencies at high or low temperatures. A cycle consisted of 24 hr at 250 F and 24 hr at −40 F. No failures occurred.

Road test evaluation in Arizona and on the endurance track at the Proving Ground were also satisfactory.

A production run was made during the 1960 model year to determine the handling characteristics of the plastic housing on the assembly line. No performance deficiencies were found. The reaction of the workers was excellent because of the much lighter weight (2 lb versus 9.5 lb) of the housing.

The acetal housing, as a painted part, has been released for the 1961 Valiant model. Until satisfactory color compounds can be obtained and adequately tested, conventional interior paints will be used.

Heater and air conditioning control switches

Push button heater and heater-air conditioning control switches made predominantly of plastic materials were developed by Chrysler engineers in 1958. These units are an excellent example of where the strength, low friction characteristics, good electrical resistance, and moldability of selected plastic materials can be combined to produce efficient and economical parts.

Fig. 3 shows an assembly drawing of the heater — air conditioning switch. The housing *V*, cover *A*, 5 push buttons *M*, and insulator *E* are molded from a heat resistant ABS compound. The four slides *G*, *H*, *S*, and *T* and the two retainers *F* are die cut from a melamine asbestos laminate. The valve *N* is a molded 70 durometer, high nitrile rubber compound which is chlorinated to reduce the frictional characteristics of this part. The remaining parts are metal terminals, springs, and contactors.

The operating principle of the switch is simple.

When a push button is engaged it moves the spring contact *K* onto the terminal *U*, thereby starting a blower motor. At the same time, it correctly posiions the vacuum valve *N* by exerting an edgewise pressure on the saw-tooth slides, causing them to move into their predetermined location. Vacuum actuators then operate the vent doors in the heater and air conditioning ducting system.

The mechanical requirements established for the unit were:

1. The operating effort required to depress each push button shall never exceed $3 \pm \frac{1}{2}$ lb.

2. The operation of each push button shall be smooth and with a positive detent.

3. Each push button of the switch assembly shall be capable of resisting five 20-lb consecutive thrusts, without damage to any of its components.

4. Each terminal shall be capable of withstanding a 40-lb pull. Minute elongation of the sealing pegs is not objectionable if the back-cover returns to its normal position when the pull force is removed.

5. The switch must withstand a durability test of 10,000 cycles under the following conditions. (One operating cycle shall consist of a forward and a return stroke of one push button. The operating thrust on each push button shall not be less than 10 lb.)

   a. 2500 cycles while operating the *Off* and *LO* or *Fresh Cool* push buttons at a rate of 6 cpm at 77 ± 3 F.

   b. 2500 cycles while operating the *Off* and *Defrost* push button at a rate of 6 cpm at 95% relative humidity.

   c. 2500 cycles while operating at *Off* and *Vent* or *Maximum Cool* push button at a rate of 6 cpm at 150 F.

   d. 2500 cycles while operating the *Off* and *Hi-Heat* push buttons at a rate of 4 cpm at −20 F.

   Upon completion of the life test, there shall be no excessive wear or breakage.

The critical factors involved in this assembly are: the strength of the plastic slides enabling them to withstand high edgewise impact; dimensional stability of the cover at temperatures up to 200 F (short duration); and toughness and strength of the housing material because the cover is fastened to the housing by means of six heat swedged pegs.

An ABS material, Cycolac, was selected as the original housing, cover, and push button material because of its toughness, good electrical properties, low friction characteristics, and cost. It was believed that the dimensional stability and strength would be adequate under operating conditions, but these properties would have to be proven by the tests outlined above.

In the heater switch the material performed very satisfactorily, but a high heat problem developed in the heater — air conditioning switch. Temperatures above 250 F resulted due to a high millivolt drop across the brass contacts with a 17 amp current. The ABS resin warped and in some cases melted, causing complete failure. This condition was corrected by replacing the brass contacts with silver coated contacts which gave a lower millivolt drop. The maximum temperature buildup under high ambient air temperatures (110 F) was only 177 F with this setup. The cover is dimensionally stable at this temperature and no further failures occurred.

In the early experimental units, some breakage was encountered in the slides when either a paper base or asbestos mat phenolic laminate was used. The selection of a melamine asbestos mat laminate completely resolved this problem.

The performance of the plastic materials in these switches has been excellent. The parts are com-

The Author

Dr. A. J. Carter

DR. A. J. CARTER joined the Rubber and Plastics Department of Chrysler Engineering in July, 1940 after receiving his Doctor's degree in Organic Chemistry at Iowa State College.

Since that time he has had considerable experience in the development and application of plastics, rubber, fabrics, cements, sealers, and friction materials in the automotive field.

In December, 1957 he was appointed to his present position of Assistant Chief Engineer in charge of Organic Materials Development.

Dr. Carter is a member of the General Materials Council and the Non-Metallic Materials Committee of the SAE.

THIS IS THE FOURTH of a series of six articles, written exclusively for SAE Journal, on "Designing with Plastics for Automotive Applications."

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pact, economical, and efficient and can be considered a very satisfactory application of plastics.

Plastic gears

Much work has been done to utilize plastics for gear application. Quiet operation and lower costs are the primary advantages. The two materials most commonly used are the phenolics and the nylons. Since nylon is thermoplastic, it may be employed successfully where comparatively light loads are involved and the temperatures are not over 250 F. Nylon may be used with or without metal inserts. Its specific advantages are low friction characteristics and ease of fabrication (injection molded).

The thermosetting characteristic of the phenolics will permit a satisfactory gear performance under greater load conditions than the nylons and at temperatures near 350 F. Gears of this material may be molded or machined from laminated stock or molded blanks.

In gear applications, the main problems are dimensional stability, strength under adverse operating conditions, and gear life as it applies to wear and fatigue. Answers to these problems can be obtained only by testing under operating conditions. Therefore, the final design is generally arrived at by a trial and error method in which several design changes are usually made before a satisfactory part is obtained. In the case of phenolic molded gears, or gear blanks, many compounds may be evaluated to find the optimum material.

An example illustrating the use of a plastic gear, where very close dimensional tolerance is required, is the rear oil pump outer rotor gear used in the Chrysler TorqueFlite transmission for 8-cyl cars. Fig. 4 shows an assembly of this plastic gear and the steel inner rotor gear in the aluminum housing. The dimensions called for are:

	Dimensions, in.	Tolerance, in.
thickness	0.4985–0.4990	0.0005
outside diameter	3.1775–3.1815	0.004
inside diameter	2.344 –2.348	0.004
Total indicator run out of flatness		0.001

The development of this plastic gear was initiated primarily to reduce the noise level and a wear condition on the aluminum housing. Since the steel gear design appeared to be satisfactory, the designers decided to duplicate it in the plastic material.

The material selected was a glass filled phenolic compound which would give maximum dimensional stability, heat resistance, and strength. After several evaluations, a ½ in. glass fiber length was finally established. This length permitted adequate flow of fibers into the tooth sections.

It was evident from the beginning that the thickness tolerance would have to be obtained by a grinding operation. The gear was ground on both sides in an upright position using a Gardner grinder. Two grinds were made, a rough grind, followed by a finish grind of 80 microin.

The initial experimental parts were molded in the laboratory in a single cavity positive die. They were very stable dimensionally and performed well in the transmission. However, when production was started in a 12-cavity mold, the parts, after machining, had a tendency to shrink as much as 0.0025 in. and run out of flatness as high as 0.004 in. when a heat test of 16 hr at 320 F was carried out. Such dimensional changes could not be tolerated because of leakage and high wear, so a normalizing procedure was established to stabilize the molded gear. When the gears were heated at 320 F for 12 hr before grinding, the specified tolerances were met.

This gear has now been successfully used for three years. Its dimensional stability and good wearing characteristics have encouraged the use of this type material in another molded transmission part, a reverse clutch stationary core in a transmission made by General Motors.

Some of the important nylon gear applications in our industry today are windshield wiper motor gears, speedometer pinion gears, and door window-lift gears. In all three cases, considerable engineering work was required to perfect these parts.

There are many other outstanding examples where plastics have performed satisfactorily as mechanical parts. Such phenolic uses as water pump impellors, distributor caps and rotors, terminal blocks, and starter brush holders are so well established that we tend to overlook these important applications. Comparable lists can be cited for the nylons, polyesters, alkyds, vinyls, butyrates, and acrylics.

As new polymers with improved physical properties become available the usage of plastics in mechanical applications will grow. The utilization of this class of material will, however, always present a challenge to the design and materials engineers because of the inherent limitations of plastics. The design and material selection must minimize these effects and offer enough desirable properties to fulfill the part requirement.

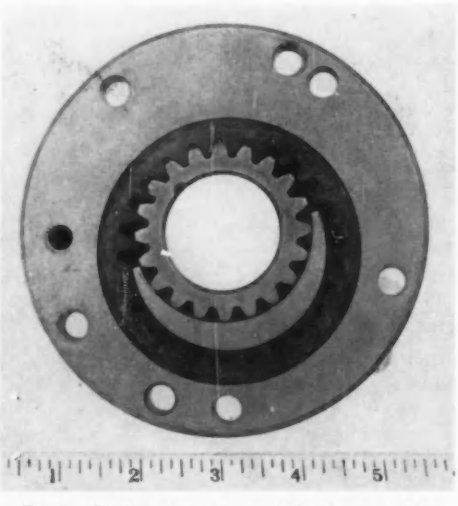

Fig. 4 — Plastic outer rotor gear in housing assembly.

Protecting against protons—

—charged particle hazards to man in space

The amount of shielding needed depends on distances from the sun, the stay-time, the solar-cycle phase, and the allowable radiation dose to space-ship crew.

Based on paper by

Donald H. Robey

Convair (Astronautics) Division, General Dynamics Corp.

PROTONS are probably the outstanding, charged-particle hazard to man in space.

These particles, which are hydrogen ions, make up about 80% of the cosmic ray influx from the galaxy (see Table 1). They also constitute 85–100% of the charged particles which stream out from the sun during solar flares. The largest flares may produce particle fluxes as high as 10,000 times the cosmic ray background for short periods of time with energies ranging from a few mev to over 1 bev.

Protons are also found, along with electrons, in regions where charged particles are trapped by planetary magnetic fields (see Table 2). In the earth's Van Allen belts the electrons have intensities high enough to produce intense short-wave, electromagnetic radiations (Bremsstrahlung) capable of easily penetrating 0.1-cm stainless steel vehicle walls.

The curves in Fig. 1 show the approximate symmetry of the first two Van Allen belts with respect to the geomagnetic axis. The trapped particle intensities tend to be greatest at the geomagnetic equator. The curve in Fig. 2 shows the variation of ionization with latitude measured by Pioneer II at an altitude of 1500 km (930 miles). At latitudes higher than about 70 deg the trapped particles appear to be negligible, although it has been rumored that they may exist directly over the Poles.

Of more importance is the fact that, over regions believed to be free of trapped particles, solar protons can stream down uninhibited by the geomagnetic field. It is believed that it is possible the particles might actually be magnetically focused and intensified over the polar caps. The auroral zones are around 68 deg geomagnetic latitude and if protons are responsible they would have to have energies of the order of 500 mev. A 1 bev proton, incident vertically, can penetrate to about 60 deg geomagnetic latitude (neglecting anomalies).

Auroral X-rays have been observed during magnetic storms at lower latitudes (56 deg geomagnetic) but they are believed to be caused by electrons from the outer Van Allen belt. The intensities are estimated to be of the order of a milli-Roentgen per hour, which is not serious.

If the crew of a space vehicle is to be protected from protons with energies as high as 500 to 1000 mev by taking advantage of the earth's magnetic field, the vehicle should not orbit above a latitude of about 60 deg geomagnetic. Also, the vehicle should

Table 1

Nucleus	Number, %	Flux Density
H (proton)	80	4000
He (alpha	19	950
Li, Be, B	1	
C, N, O		33
Na, Mg, Al, Si		6
S, A, Ca		2
Fe		1
Magnetic rigidity > 1.5 BV		

ABOUT 80% OF THE COSMIC RAY INFLUX FROM THE GALAXY of which the sun is a part, consists of charged particles (protons), as this table shows.

The table summarizes some of the available data about the nature of cosmic ray nuclei, the per cent by number, and the corresponding flux density, taking iron as unity.

The heavier primaries like iron are seen to be scarce. Because of their larger size, however, the possible damage they might do to the human brain is a matter of conjecture.

Some radiobiologists think that a single, heavy primary particle, like iron, could put a man out of action if it struck the brain in a critical area.

PROTONS ARE ALSO FOUND, ALONG WITH ELECTRONS, in regions like the earth's Van Allen Belt, where charged particles are trapped by planetary magnetic fields.

In the earth's Van Allen Belt, this table shows, the electrons have intensities high enough to produce intense short-wave, electromagnetic radiations . . . capable of penetrating 0.1 – cm stainless steel walls.

The table shows also the compositions and intensities of the inner and outer belts of Van Allen radiation shown in Fig. 1.

Table 2

	Inner Belt (maximum Intensity -- 3600 km)	
Electrons,	$E > 40$ kev – 2×10^9/cm²/sec/sterad	Unidirectional intensity
	$E > 600$ kev – 10^7/cm²/sec/sterad	
Protons,	$E > 40$ mev – 2×10^4/cm²/sec	Omnidirectional intensity
	Outer Belt (maximum intensity – 23,000 km)	
Electrons,	$E > 40$ kev – 10^{11}/cm²/sec	Omnidirectional intensity
	$E > 200$ kev – 10^8/cm²/sec	
Protons,	$E > 60$ mev – 10^2/cm²/sec	

protecting against protons

. . . continued

remain below an altitude of approximately 700 km (434 miles) where the flux densities in the inner belt are still low. Since the geomagnetic poles are not on the earth's axis of rotation, the geomagnetic latitudes tend to oscillate with respect to an observer on an Earth satellite. The changes in latitude are about ± 14 deg N as compared to ± 22 deg S. Thus, to be on the safe side the vehicle should remain below geographic latitudes of 38 deg N.

Additional protection can be had by remaining close to the geomagnetic equator. However, solar flares with intensities high enough to be hazardous above energies of 1 bev are rare, occurring about once every three or four years. The largest flare on record would not be lethal if all protons with energies below 1 bev were removed.

The protons in the inner Van Allen belt make shielding mandatory for sustained manned flight in this region. A calculation was made to determine the order of magnitude of shield weight required for flight in the most intense part of the inner belt. The results are shown in Fig. 3. In calculating the rem rate, an RBE of 1 was used for protons with energies higher than 40 mev, and an RBE of 2 for protons with lower energies. Carbon was used for the spherical shield, which has an inside radius of 122 cm (4 ft). A shield of this type, weighing 45,000 lb, would keep the radiation dose rate down to the order of 0.5–1.0 rem/hr.

In the outer belt, the Bremsstrahlung from trapped electrons may require an additional inside lining of iron or lead probably of the order of a few millimeters.

The possible presence of solar flare particles over the polar caps makes a polar orbit for the manned space station somewhat undesirable, especially near the top of the solar cycle. In these regions the dose rates can reach hundreds of rem/hr with particles in the mev to bev energy range. The dose rate following a solar flare from protons of energies of 100 mev or more over Minneapolis (geomagnetic latitude about 59 deg, geographic latitude about 45 deg) was found to be 0.1 rem/hr. The measurements were made from high-altitude balloons. This dose rate is probably the right order of magnitude for the dose rate to a crew of a low-altitude space station, with 2.9 cm of carbon shielding, flying over the latitude of Minneapolis. It would be higher at higher altitudes or latitudes. Over the polar caps the dose rates can be hundreds of rem/hr for periods for many hours.

The solar flare hazard is considered to be especially serious for travel in interplanetary or cislunar space. Two of the largest flares in recent years were analyzed and found to be lethal to unshielded man at the Earth's distance from the Sun.

The flare of May 10, 1959, which was the largest of the year, would have required a closed shield with about 50 g/cm² of carbon between the occupants and external space to reduce the radiation dose to a few rem. A spherical shield of this type with an inside radius of 90 cm (3 ft) weighs about 14,000 lb.

The most hazardous flare on record, the flare of Feb. 23, 1956, probably would have required ten times as much carbon to do the same job.

However, rather than shield for individual flares, the problem is to integrate the effects of flares of all sizes, along with the expected cosmic ray background. If the area and light intensities of flares turn out to be important criteria for particle production, then the subflares may be as important over a year's time as the year's total of large flares.

A study program of this kind is being carried out at the present time. The amount of shielding required will depend on the distances from the Sun, the stay time, the phase of the solar cycle, and the allowable radiation dose to the crew.

This article is based on part of an Astronautic Symposium developed jointly by SAE and the Air Force Office of Scientific Research. The Symposium is available only as a book, titled **"Vistas in Astronautics — 1960."** To order, turn to p. 6.

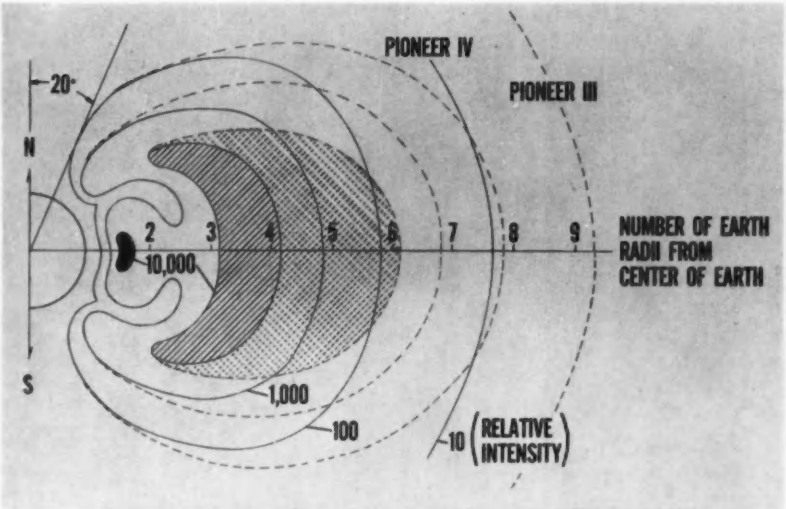

Fig. 1 — **THE VAN ALLEN TRAPPED-PARTICLE BELTS** are shown here in a rough cross-sectional view. The contours show regions of constant intensity. Within the shaded region, the counts reach 25,000 per sec.

Compression of the Earth's magnetic field by fast-moving plasmas from the Sun can raise the flux densities and increase the trapped-particle intensites. Resulting temperature increases presumably cause expansions in the outer plasma.

A possible effect of this type is illustrated in this chart where the outer belt is undergoing contraction (cooling). . . . The difference in altitude (over the magnetic equator) is equal to about two Earth radii, according to data received from Pioneer III and Pioneer IV. . . . A solar "m" region event on Feb. 25, 1959 is believed to have caused a considerable extension of the radial limits of the outer zone.

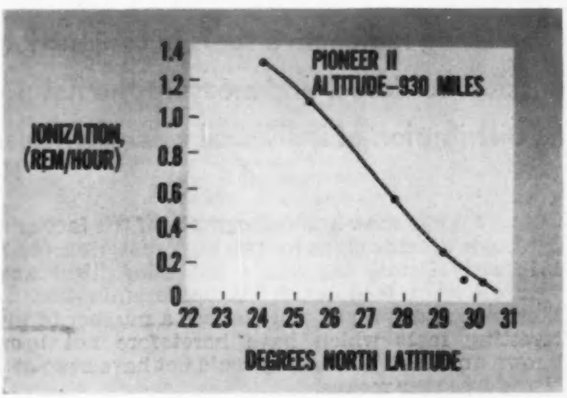

Fig. 2 — **VARIATION OF IONIZATION WITH LATITUDE** measured by Pioneer II at an altitude of 930 miles is shown here. . . . At latitudes higher than about 70 deg, the trapped-particles appear to be negligible. Some think, however, that trapped-particles may exist directly over the poles.

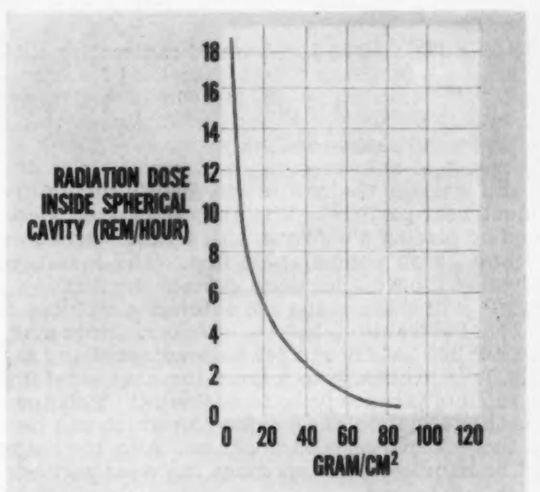

Fig. 3 — **PROTONS IN THE INNER VAN ALLEN BELT** make shielding mandatory for sustained manned flight in this region. This chart shows the results of a calculation made to determine the order of magnitude of weight for a spherical carbon radiation shield required for flight in the most intense part of the inner belt.

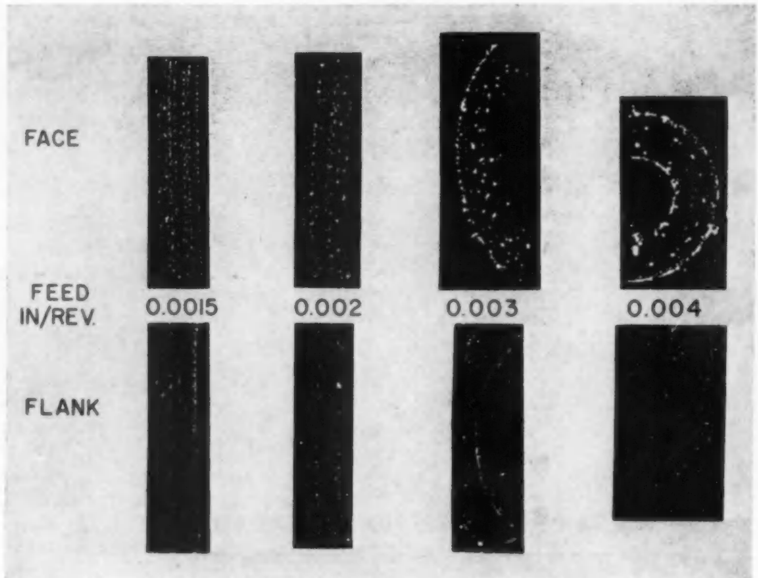

Fig. 1 — Autoradiographs of the face and flank side of lathe chips of SAE 1018 carbon steel — 30 cut speed.

Radioisotopes spot tool wear

Using radioactive tools in conjunction with autoradiographs, it is possible to detect and measure the nature and distribution of individual wear particles.

Based on paper by

M. Paliobagis and E. J. Krabacher

The Cincinnati Milling Machine Co.

RADIOISOTOPES in the form of radioactive cutting tools provide a powerful method for investigating the nature and mechanism of tool wear. This article discusses recent special studies made possible by radioactive tooling.

By means of autoradiographs it is possible to detect and measure the nature and distribution of individual wear particles. The autoradiograph is obtained by placing a chip cut with a radioactive tool in contact with photographic film. The radiation emitted by the wear particles exposes the film.

In preparing the chips for autoradiographing, a number of precautions have to be taken. Since most chips are not flat enough for autoradiographing directly, it is necessary to anneal the chip carefully in a vacuum furnace prior to flattening. This prevents the formation of an oxide film which can disturb the wear particles on the chips. Also, the chips must be handled with care since the wear particles are easily detached.

Figs. 1 and 2 show autoradiographs of the face and flank side of lathe chips for two work materials (SAE 1018 carbon steel and SAE 81B45 alloy steel) and four different feed rates. Visual examination of these autoradiographs will disclose a number of interesting facts which have heretofore not been known and in all probability could not have been disclosed by other means.

For SAE 1018, it can be seen that the concentration of wear particles per unit area decreases rapidly with increasing feed, while the physical size of the wear particles increases. At the 0.0015 feed rate the wear particles must be extremely small and densely distributed in order to give such a uniform exposure of the photographic film. At the 0.004 feed the wear particles have become discrete and are much larger in size.

For the SAE 81B45 we have the same trends, density of wear particles decreases with feed, while the size of the wear particles increases. However, the absolute magnitude of the concentration and physical size of the wear particles are quite different. While it isn't possible here to give a complete interpretation of these autoradiographs, it is clearly evident that the use of such techniques can shed much

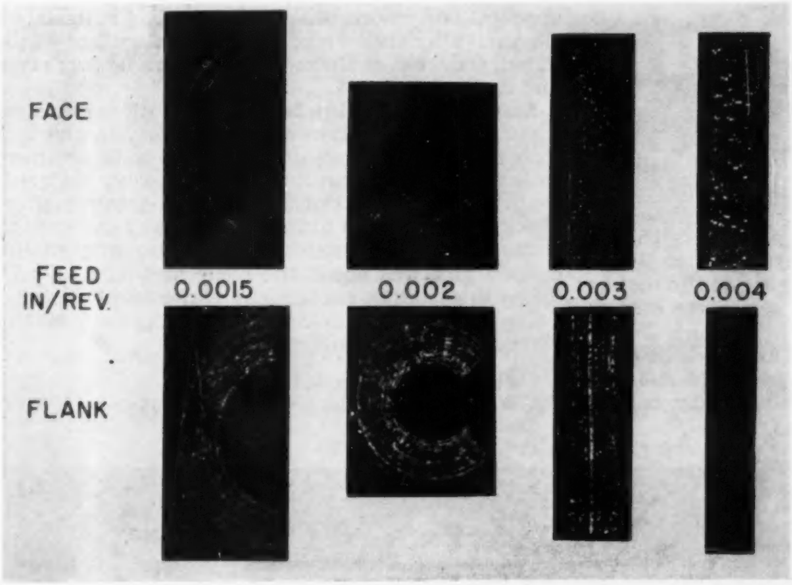

Fig. 2 — Autoradiographs of the face and flank side of lathe chips of SAE 81B45 alloy steel — 30 cut speed.

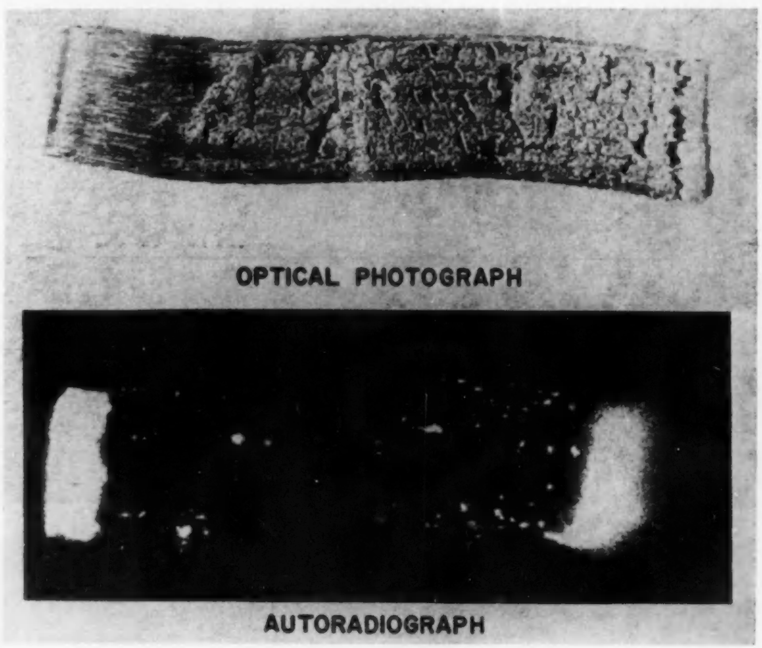

Fig. 3 — Optical photograph and autoradiograph of the face side of a milling chip. Note the high concentration of wear particles at the left caused by impact of the tool entering the workpiece. The high concentration at the right results from the presence of the built-up edge on the tool.

needed information on the nature and mechanism of tool wear.

The application of autoradiographic techniques to the study of milling chips has yielded interesting information on the role of the built-up edge as related to tool wear. Fig. 3 shows an optical photograph and an autoradiograph of the face side of a milling chip. At the left is that portion of the chip that was formed when the cutter first entered the work material. The high concentration of wear particles is the result of impact as the tool enters the workpiece. This effect of impact can also be seen in the autoradiograph of the workpiece surface, Fig. 4.

The top of Fig. 4 shows the high concentration of wear particles that results each time the cutter enters the workpiece. As the cutter continues through the workpiece, the wear particles become less densely and more uniformly distributed. On the right hand side of the chip autoradiograph (Fig. 3), we again have an area of high concentration of wear particles. This high concentration results from the presence of the built-up edge on the tool, for as the tool leaves

Radioisotopes spot tool wear

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the workpiece the built-up edge remains with the chip, but in being removed from the tool it plucks material from the tool. This plucking of the tool as the built-up edge is sloughed off can occur during any stage of the cut.

Fig. 5 shows this sloughing off of the built-up edge along the whole length of the chip. Thus, the importance of the presence of the built-up edge can be appreciated. If this plucking of the tool material by the sloughing built-up edge can be reduced or eliminated, the wear of the cutting tool can be decreased and its life increased.

Another study of the transfer of work material to a cutting tool used a variation of the radioactive tool wear method. In place of radioactive tools, ordinary tools were used to cut a radioactive work material, copper. The amount of transfer was determined by making quantitative autoradiographs of the cutting tools. Here, the amount of metal transfer to the cutting face was about the same as would be produced in a sliding experiment under similar conditions. The use of lubricants drastically reduced the amount of metal transfer.

To Order Paper No. 181A . . .
from which material for this article was drawn, see p. 6.

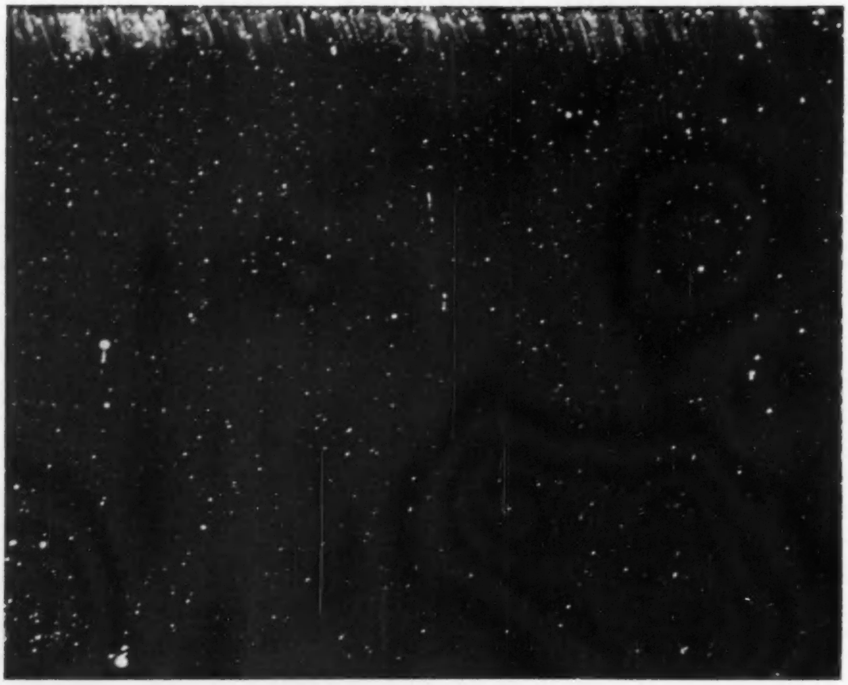

Fig. 4 — Autoradiograph of workpiece surface. The top of the picture shows the high concentration of wear particles that results each time the cutter enters the workpiece, the wear particles become less densely and more uniformly distributed.

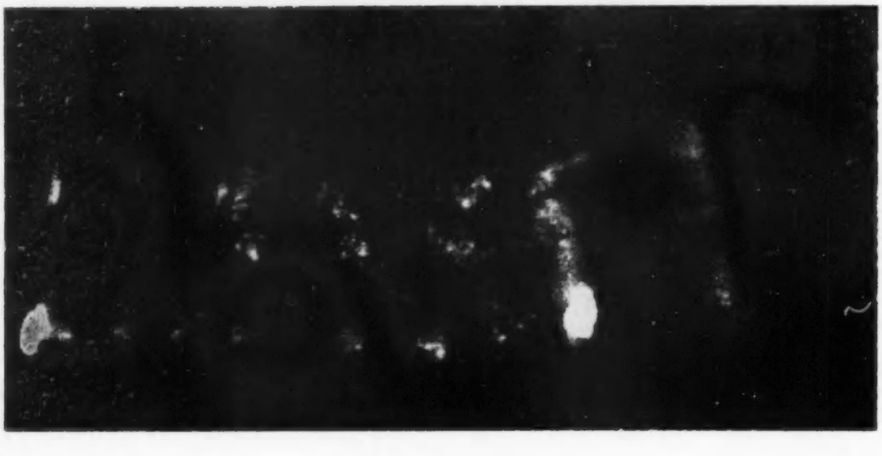

Fig. 5 — Sloughing off of the built-up edge along the length of a chip. If plucking of the tool material by the sloughing built-up edge can be reduced or eliminated, the wear of the cutting tool can be decreased and its life increased.

7 design rules help reduce GEAR NOISE

Based on paper by

William D. Route

Chevrolet Engineering Center, GMC

HERE ARE some design rules that may prove helpful in controlling noise of gears, such as those used in automotive transmissions:

1. Use a total contact ratio greater than 2.5, with a least involute contact ratio of not less than 1.2.

2. Select the finest pitch and the lowest pressure angle consistent with strength requirements. Avoid tooth action close to the base circle.

3. Use the highest helix angle consistent with size of thrust loads and face width.

4. Favor a long angle of recess and short approach.

5. Select an even number of pinions and spacing, so half of them engage halfway in the interval of engagement between the others.

6. Provide backlash, dimensional control of mounting, or floating elements to avoid tight mesh.

7. Modify lead and involute to compensate for tooth deflections and dimensional errors.

Most of these rules can be justified by hypothesis and popular usage, although few can be supported by well-documented data.

1. Involute and helical contact ratio

Involute and helical contact ratios are a measure of the number of teeth in contact, or:

$$m_p = \frac{Z}{P_b} \tag{1}$$

$$m_F = \frac{F \tan \psi_b}{P_b} \tag{2}$$

where:

m_p = Involute contact ratio
Z = Length of line of action
P_b = Base pitch
m_F = Helical contact ratio
F = Active face width
ψ_b = Base helix angle

It can be seen from equation (1) that involute contact ratio is the length of the line of action per

GEAR NOISE

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unit base pitch. The helical contact ratio, equation (2), is the amount which the length of the line of contact has been extended per unit base pitch. To arrive at a total number of teeth in contact per unit pitch it is customary to add these two ratios.

The integers which bound this sum represent the numbers of teeth which alternately have some portion of their length in engagement. However, from Fig. 1 it can be seen that, at the instances when a tooth is in contact at point A or B, the points of entrance and exist to the field of contact, only a point on the tooth length is engaged. Therefore, the simple sum of the contact ratios does not give a very true appraisal of the load distribution. A better measure of the teeth in contact is obtained by dividing the total length of the contact lines in the field of action (Fig. 1) by the cosecant of the base helix angle (ψ_b) and the base pitch (P_b). Since the total length of the contact lines is a discontinuous function, its mathematical statement is a number of expressions, the selection of which depends upon the point along the line of action to be considered and the relative value of m_p and m_F.

In Fig. 2 is plotted number of teeth in contact (or the total length of contact lines divided by $P_b \csc \psi_b$) against the length of the line of action Z. The largest variation in the number of teeth occurs when the contact ratios are:

$$m_p - n = m_F - K = 0.5$$

and the number of teeth in contact (or the total length of the contact line) remains constant if either m_p or m_F is an integer. The table in Fig. 2 lists the maximum and minimum values of teeth in contact as well as the sum of m_p and m_F. It will be noted that, in the range of involute and helical contact ratios used in automotive gears, these sums differ considerably.

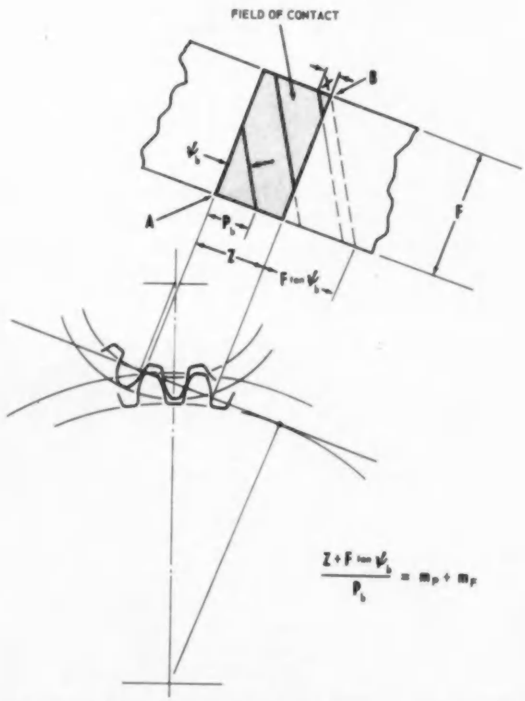


Fig. 1 — Helical gear tooth action — note that x is the distance that tooth element A travels before the next abrupt change in tooth load occurs (that is, when B leaves engagement).

Contact ratio, or number of teeth in contact, is considered important in noise control because of its role in tooth load fluctuation. For example, if we neglect the transient load fluctuations due to the teeth entering engagement at some velocity, we can draw an idealized picture of how tooth load fluctuation is reduced with increasing overlap. Fig. 3 is such a picture.

The slope of the load lines is due to the increase in tooth stiffness as contact moves toward the pitch point and decrease in tooth stiffness as the contact moves away from the pitch point. (The absolute values of load division are not meant to be significant. The values shown are exaggerated because they were obtained by calculation assuming the teeth deflect as cantilevers of uniform section.)

It will be noted that the load fluctuation at tooth contact frequency is a function of overlap and if the overlap can be sufficiently extended these load fluctuations can be made smaller. It is reasonable to expect that this reduction in load fluctuation will result in lower amplitude elastic displacements at tooth contact frequency and less noise potential to the vehicle.

Most production transmission designs have a total contact ratio (the sum of m_p and m_F) between two and three. An involute contact ratio of 1.4 is considered desirable. The involute contact ratio when all tolerances accumulate in the direction for the least ratio should not be less than 1.2.

There is more variation among contemporary designs in helical contact ratios; ratios between 1.4 and 1.7 are most common.

2. Pitch and pressure angle

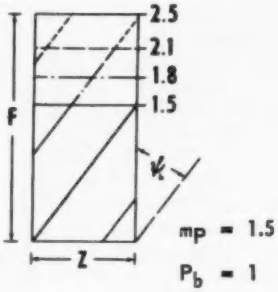
Pitch — It is generally agreed that fine-pitch gears are less noise sensitive than coarse-pitch gears, and it is therefore desirable to select the finest pitch consistent with the bending strength requirements.

In automotive passenger-car transmission planetary applications, 12 diametral pitch is considered a very coarse pitch and 20 is the highest found in production.

For the countershaft transmissions, where the tooth loads are usually higher, the pitches are coarser. Eight to eleven pitch is used on the constant-mesh gears. Six to eight pitch is used on sliding gears.

The reason usually proposed for the quieter action of the fine-pitch gears is that involute and helical contact ratio are both linear functions of diametral pitch; therefore, it is easier to obtain adequate teeth in contact if the pitch is small. More teeth in contact means lower loads per tooth.

But for the same contact ratios, fine-pitch gears



MP + MF	LENGTH OF CONTACT LINES ÷ P_b csc ψ_B MAX.	MIN.
2.7	1.9	1.7
3.0	2.5	2.0
3.3	2.8	2.6
3.5	3.0	3.0
3.6	3.2	3.1
4.0	4.0	3.5

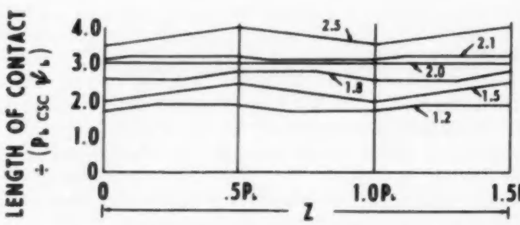


Fig. 2 — Length of contact versus Z (length of line of action).

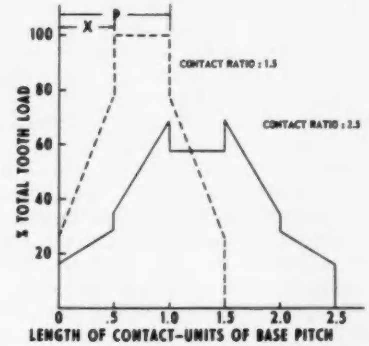


Fig. 3 — Tooth load versus length of contact.

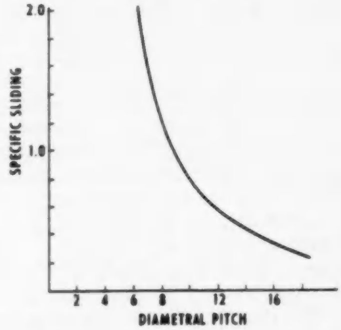


Fig. 4 — Specific sliding versus diametral pitch.

seem to be inherently quieter. The explanation for this lies, it is thought, in the fact that, for the same contact ratio, the tooth action remains closer to the pitch circle, with a consequent reduction in specific sliding and, therefore, involute sensitivity. Specific sliding versus diametral pitch for a particular gearset is shown in Fig. 4. The increasing distance of the start of action from the base circle is shown in Fig. 5. All the other features of this gearset have remained constant, that is, involute overlap, center distance, and base circle diameter. The addenda (as shown in Fig. 6) have decreased with increasing pitch to keep the contact ratio unchanged.

There are those who also attach some significance to the fact that, for the same contact ratio, the relative sliding is less with the finer pitch. For our sample gearset, the sliding as a function of diametral pitch has the same general character as that shown for specific sliding versus diametral pitch, as shown in Fig. 4. If sliding is a significant consideration, then it must be presumed that the sliding interval, at tooth contact frequency, is capable of developing forces which generate gear sound.

It is frequently assumed that the rate of the finer-pitch gear is lower than that of the coarse-pitch gears. In Fig. 7 the relative tooth stiffness of the gearset mentioned is plotted as a function of diametral pitch. Note that the 15-diametral-pitch gear is actually stiffer than the 7-pitch gear. The reason is that the tooth whole depth is decreasing (for a constant contact ratio) faster than the tooth thickness is decreasing. It is thought that this stiffening effect does not help noise control, it is probably a negative factor. Therefore, it must be concluded that the quieter action of the finer pitches is a result of increased contact ratio with reduced specific sliding and perhaps relative sliding.

Pressure Angle — Normal pressure angles in contemporary planetary gearsets range between 18 and 24 deg. In this range, involute contact ratio decreases rather rapidly with increasing pressure

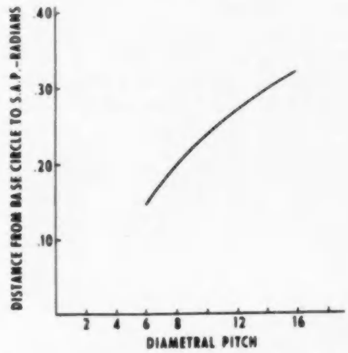


Fig. 5 — Distance from base circle to start of action versus diametral pitch.

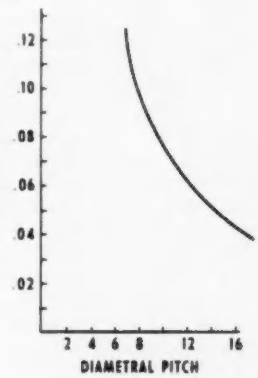


Fig. 6 — Addenda versus diametral pitch.

GEAR NOISE

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angle. Therefore, from the standpoint of achieving a large or adequate contact ratio, the lower pressure angles are preferred.

Pressure angles on countershaft transmissions are, on the average, slightly lower than on planetaries in spite of the higher tooth loads; the average range is about 16–18 on constant-mesh gears and higher on sliding gears. The use of lower pressure angles in countershaft arrangements is probably a design compensation for coarser pitches and the consequent loss of contact ratio. In other instances the lower pressure angles have been used to avoid pointed teeth resulting from large helix angles.

From the standpoint of noise reduction, the low pressure angle is attractive because it reduces tooth stiffness. At this writing, we can only say that reduction of tooth rate is directionally desirable because it will reduce dynamic transient loads encountered at the start and end of engagement. Also, lower pressure angles will minimize the load variations produced by dimensional errors.

The limiting considerations to a low pressure angle are (1) proximity of the start of action to the base circle (high values of specific sliding) and (2) bending and compressive stresses which increase with decreasing pressure angles.

When considering the limitations on the proximity of the start of action to the base circle, a rule might be, "considering all dimensional tolerances, the lowest point of contact will be a radial distance of 0.010 above the base circle." After quoting the rule, we must hasten to say that several successful production designs with relatively fine pitch pinions extend the contact to within 0.005 of the base circle. Some designers follow a rule that the lowest point of contact should not be closer than 8–10 deg of base circle roll.

3. Helix angle

The overlap obtained as a consequence of helix contributes much to the relatively low noise level of automotive gears. Also, as noted in Fig. 1, the helical tooth begins and ends contact as a point and spreads its contact diagonally across the face of the gear. Therefore, its effective rates at the start and end of engagement are reduced and, like a lower pressure angle, will probably experience less load fluctuation as a consequence of dynamic transients at start and end of engagement.

Helix angles of planetary transmissions vary from 15 to 23 deg, while for countershaft transmissions, helix angles as high as 45 deg have been produced. The limitation on helix angle selection is generally the size of the thrust loads which can be tolerated. Here again the choice of a planetary arrangement which affords the smallest tooth loads for a given input will permit the use of higher helix angles.

The other quantity in determining helical contact ratio is face width and here the designer is limited by space and dimensional tolerance.

There is little question but what helical contact ratios much higher than those currently used in automotive gears would be effective in reducing noise. But in fact, to use this increased overlap might well require better accuracy than is currently found in our gears. This is an area where precise knowledge of the elastic rate of the gear teeth and the involute, lead and spacing accuracy capabilities of the production equipment should be made available to the designer. With this information a study can be instituted to arrive at the usable helical ratio.

4. Angles of approach and recess

A design feature considered by some authorities as important in obtaining quiet tooth action is the proportioning of addenda, so that the angle of approach is small and the angle of recess large. In a planetary arrangement, a large recess between the sun and pinion will tend to produce a large approach between the pinion and ring unless the designer resorts to a different (lower) operating pressure angle between the pinion and ring than between the sun and pinion. For countershaft transmissions, this arrangement, of a small approach and large recess, can be accomplished directly.

For spur gears the argument for this kind of modification is fairly clear. The tooth action is quieter during the angle of recess because of the direction of the friction forces during this interval. The normal tooth force is a function of the friction coefficient between the teeth and can be expressed as follows:

$$W_n = \frac{T}{R_b(1 - \mu \tan \phi)}$$

in the angle of approach

$$W_n = \frac{T}{R_b(1 - \mu \tan \varphi)}$$

in the angle of recess

where:

W_n = Normal tooth force
T = Input torque, lb-in.
R_b = Base circle radius, in.
ϕ = Pressure angle at radius contact
μ = Friction coefficient

In the angle of approach, the normal tooth load increases with friction; while in the angle of recess, the normal load decreases. Thus, during approach, the teeth are pressed harder together, with increasing friction, and if the friction is sufficiently high it could lock up the gearset (when $\mu = \text{ctn}\,\varphi$). Under such extreme conditions, it might be expected that the tooth action would be noisy.

For helical gears this argument becomes less impressive because the line of contact along the length of the tooth is oblique. Therefore, on the same tooth, sliding can occur on both sides of the pitch diameter simultaneously. For the small helical contact ratios encountered in automotive gears, the argument may still have some validity.

However, it is unlikely that gear teeth with the finish of automotive gears, operating in a well-lubricated environment, experience friction forces of

sufficient to avoid tight mesh (contact on two sides

significant value. For example, if the friction coefficient were 0.1 (which would be a pretty sticky gearset) the normal force variation would be less than 10%.

On the other hand, poor surface finishes which result in metal shear during the sliding action might well make gears noisy, but they should respond to improved surface finish faster than addendum modifications.

5. Pinion spacing

The pinions of the planetary gearset are normally spaced equally about the carrier if their number is odd, and located diametrically opposite each other when their number is even. This spacing is attractive because the net bearing load on the carrier is zero.

The choice of numbers of teeth of the planetary and pinion spacing about the carrier determines whether the pinions will simultaneously start contact or whether they will alternately start contact with respect to one another. Pinion spacing which provides alternate tooth engagement can sometimes be helpful in the reduction of pure gear noise. It is helpful, particularly where there is an even number of pinions, to choose their spacing so that half of them will start engagement precisely half way between the interval of engagement of the other pinions.

That this type of pinion spacing is helpful in noise control is a matter of record, but why a matter of hypothesis. Direct noise, that is, noise radiating directly from the gear bodies themselves, is not heard in the vehicle. The noise heard in the body cavity is radiating from surrounding panels, which are forced or resonating at tooth contact frequency. When the disturbance from each of the pinions is identical in form (same frequency components) and the path of the disturbances from each pinion to a point of convergence is of equal length, the resulting displacement is the sum of the disturbances from each of the pinions. Thus, the resultant displacement at the sounding structure is in magnitude something greater than the disturbance from a single pinion. However, if the sinusoidal strain waves from each of the pinions are out of phase, they may tend to cancel each other at the point of convergence or at least produce a resultant less than the algebraic sum of disturbances of the several pinions. Therefore, to have pinion spacing effective in noise reduction, it would seem that two conditions must be present: (1) the disturbance from each pinion must be identical in wave length and (2) the path of each pinion disturbance to a point of convergence must be of equal length. The first condition can easily be verified by a sound analysis, but the second requirement is difficult to establish because the path of the vibrations is completely unknown and usually a matter of speculation.

Still another consideration is the simultaneous or alternate tooth engagement of a single portion at both of the geared elements (in other words, to

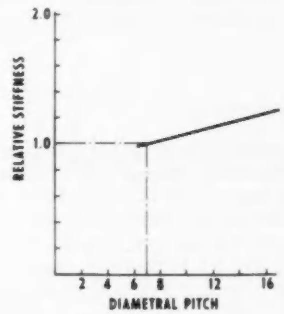

Fig. 7 — Stiffness versus diametral pitch.

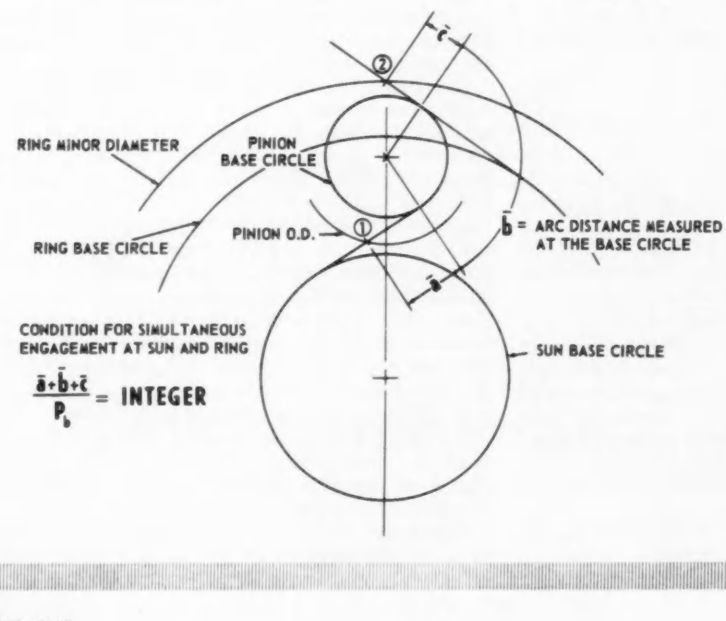

Fig. 8 — Condition for simultaneous engagement at sun and ring gears.

have a pinion engaging a tooth on the sun and another on the ring simultaneously or alternately). This also requires that a rather special relationship be held between the gear elements. Thus, in Fig. 8, the base circles and lines of action of the sun, pinion, and ring are shown (for the case of the sun driving). Contact between the sun and pinion starts at point (1). For simultaneous engagement, the distance $\overline{a}+\overline{b}+\overline{c}$ must be an integral multiple of the base pitch. This would require a special selection of pinion OD and ring ID once the base circle diameters and base pitch have been chosen.

The effect of this latter type of tooth engagement phasing is, as far as we know, an untried area and indeed the tolerances that must be accepted on the diameters will preclude a precise arrangement. For random tooth engagement, this arrangement might be developed.

6. Shaft and support design

Countershaft Transmission Arrangements — The objective in shaft and bearing design for countershaft transmission arrangements is to obtain the highest degree of stiffness permitted by the size and weight and limitations of the transmission. Under load, the gears of the countershaft train are not only deflected radially, which results in reduced contact ratio, but they are tilted due to the slope of the deflected shaft. This tilting produces end loading and loss of helical contact ratio. This loss of total contact ratio can be a major contributor to noise.

It is helpful in the initial design to study shaft deflection and slope analytically by such techniques as the conjugate beam method. Attention is given to adequate case ribbing and cover design for maximum stiffness. Under maximum engine torque, calculated deflections and slopes of 0.005 in. deflection and 0.002 radians slope in the plane of the centers are representative values. Limiting values of deflection should be established by the minimum contact ratio available.

In the development period, gear bearing pattern studies, static deflection studies, and stresscoat techniques are helpful in measuring the deflections of the assembly and altering for maximum stiffness.

The practice of using some lead modification in the form of crown on one of the two gears of each train to eliminate the end loading that will occur as the consequence of any deflection is now almost universal. Since this crowning in itself reduces helical overlap, the least possible is the most desirable. The least possible is defined by the amount of deflection permitted in the design stiffness.

In countershaft arrangements, the deflections are generally in the direction to increase backlash. Therefore, the backlash specification need only be sufficient to avoid tight mesh (contact on two sides of the gear tooth) resulting from dimensional tolerances such as tooth spacing and thickness, center distance, and outside diameters.

Excessive backlash has no effect on noise but is objectionable for other considerations such as gear rattle when unloaded, driveline noise during torque reversals, and simple loss in beam strength.

Backlash values are as variant as the number of designs but falls in the range of 0.004–0.011 measured on nominal centers.

Planetary Gear Arrangements — In planetary gearing, the radial loads on the supporting structure due to torque loads are zero if the elements are precisely concentric (and, of course, if the pinions are equally spaced or diametrically opposite). Therefore, the primary objective is to achieve the very best cencentricity between elements. Short of absolute concentricity, what must be achieved is a mounting system that prevents eccentricities between the elements greater than the center distance shift permitted by reasonable backlash in the gears.

For example, when the eccentricities enforced by the mountings between the sun gear and carrier (and the argument holds also for the ring gear and carrier) exceed the center distance shift allowed by the backlash of gears, the mounting system is required to deflect under forces provided by the gear teeth operating in tight mesh. These radial forces are a function of the elastic rate of the mounting structure and are of a magnitude to be destructive and exceedingly noisy. To reduce these forces requires one of the following:

(a) Improve the concentricity controls.
(b) Increase the backlash.
(c) Allow the geared elements to "float" with respect to the carrier and its pinion.

So-called "floating" gears are generally driven through coupling devices like the Oldham coupling or loose fitting splines or lugs. Under load, depending upon their design, all of these devices offer some constraint to a change in position. Whether they ever permit the geared members to center themselves is doubtful, because the centering forces acting on a true "floating" gear are small if indeed they exist at all.

For example, consider the free body diagram of a sum gear which is eccentric to the carrier center as shown in Fig. 9a. If the tooth loads $W_{n1\text{-}2\text{-}3}$ are all considered equal and their directions are along the lines of action, the summation of these forces is not zero but indicates the necessity for another force (force 0–5 in the polygon of forces in Fig. 9b), which must come from a further shift of the sun gear to run eccentric until it is in tight mesh. However, the tooth loads are not all equal for this eccentric condition. The tooth load at 0–1 will be less than at 2–3 or 3–4 because the "spread center" here has reduced the rate of the gear teeth and similarly increased it on the closed center condition occurring at 2–3 and 4–5. This unequal loading is in the direction to make 0–5 approach zero and indeed reverse its direction so that the system might become stable. A free body study of the ring indicates a similar load stability relationship applies there.

Whatever direction this "unbalance force" takes, it is by necessity small so that "floating" gears can effectively eliminate interference loads.

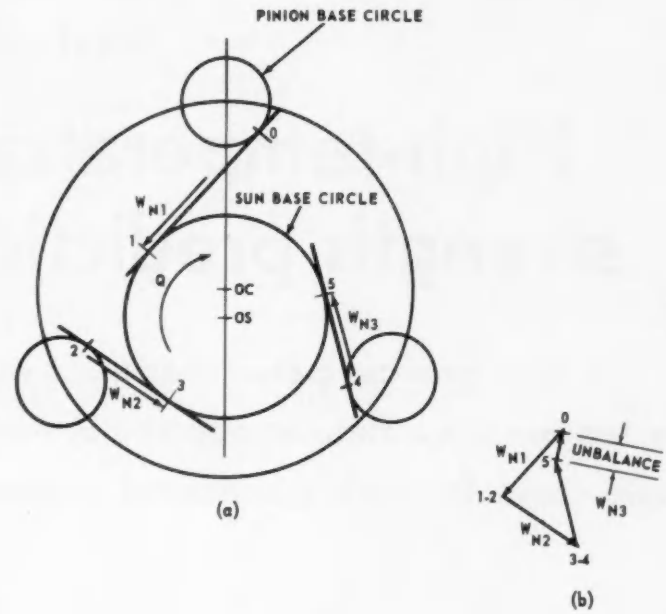

Fig. 9 — (a) Schematic of structural forces; (b) vector diagram of structural requirements.

7. Tolerances and modification

A most important final consideration in the design of transmission gears is the tolerances to be assigned to the gear-tooth dimensions. These tolerances are likely to undergo continual revision while the noise sensitivity of the design is being established by the production of statistically significant quantities of gears. Listed below are representative tolerances for this class of gearing, which may be used as starting tolerances:

1. Lead — external gears:
   (a) ± 0.0005 in. per in. of face width.
   (b) Lead variation between any two charted teeth not to exceed 0.001 in. per in. of face width.

   Lead — internal gears:
   (a) ± 0.001 in. per in. of face width.
   (b) Variation between any two teeth not to exceed 0.002 in.
2. Involute:
   (a) + 0.0002 in.
   − 0.0003 in.
   (b) Variation of involute form between any two teeth not to exceed 0.001 in.
3. Spacing:
   (a) Maximum tooth to tooth index error 0.0005 in.
   (b) Maximum indexing error 0.001 in.
4. P. D. runout — 0.001 in.

It is generally not the practice in automotive transmission gearing to use a straight, unmodified involute. Usually, the driver, or in the case of planetary trains, the pinion, has a "crowned involute." That is, looking at the involute checker trace of the profile, the involute is minus at the tip and at the root. With an unmodified mating tooth, this will produce a delayed entry and an early exit.

Involute modification may be thought of as tapered backlash. Thus, at the point of tooth engagement with no modification there may be interference because of deflection or spacing error. The modification provides some clearance at this point of theoretical start of engagement to avoid this interference. As the theoretical point of contact now moves down the line of action, this backlash tapers to zero clearance and the load is applied to the tooth at a rate depending upon the modification taper and the elastic rate of the gear teeth. It is desirable that this taper or modification washout before the next abrupt change in tooth load, or before the distance x shown in Fig. 1.

The correct amount of involute modification is best achieved as a development project, proceeding gradually from a straight involute. Such a development project involves noise measurements of a large enough sample of gearsets to insure that a real change has been accomplished by the modification.

To Order Paper No. 208E . . .
from which material for this article was drawn, see p. 6.

Short cut to

High-temperature strength prediction

A rate process parameter boils down complex temperature-time exposures of metals to a simple curve for each mechanical property.

Based on paper by

D. M. Badger and C. D. Brownfield

Norair Division, Northrop Corp.

MASSIVE CUTS in test and design time are possible for high-temperature-deterioration-of-strength problems through the use of a temperature-time parameter. This parameter, $T(C + \log t)$, has already been proved for aluminum alloy 7075-T6, and early results indicate it should be applicable for other hardened aluminums and steels.

Basically, the parameter correlates with the loss of physical properties, such as tensile, compression, and shear strength, resulting from accumulated exposures to high-temperature environments. The cumulative effect, when expressed in terms of the parameter, is independent of the way the temperature-time history was developed, thus making the analysis of a missile or aircraft structure relatively simple, despite a complex series of missions at high temperatures.

Stress applied during exposures may also accelerate permanent strength reduction, particularly if creep strain should occur. Tests show that this effect is small if the inelastic strain is small and that the effect can be calculated if the amount of inelastic strain is known.

One simplifying factor in the cumulative damage analysis is that the strength properties can usually be determined from short-time loadings. This makes performance prediction easier than for progressive failures, such as creep rupture, under complex exposure conditions.

Results for aluminum alloy 7075-T6

Overall mechanical performance can be boiled down to a couple of dimensionless ratio curves. However, several steps were taken in verifying and producing such composite results.

Single exposures

The alloy was first tested under several different temperature-time combinations, each combination being an individual test. Tensile strengths for room temperature and 300 F were then plotted versus θ, $T(C + \log t)$, as shown in Fig. 1. The different combinations produced a smooth curve.

This result bears out the theory behind the parameter θ, which is derived from the rate process equation. Theoretically, there should be an infinite number of temperature-time combinations that would give the same strength after exposure to high temperature. (A high temperature for a short time should give the same precipitation or over-aging as a lower temperature for a longer time.)

Multiple exposures

The next step proved that the alloy could be exposed to several temperature-time extreme conditions in sequence and still produce the same strength-θ relationship, provided the θ's were added. This result is shown in Fig. 2 with the solid line a replot of the single-exposure data. During the tests, specimens were exposed up to 10 different temperature-time conditions. Some sequences had single-direction temperature trends, increasing or decreasing, and others had well mixed exposure conditions.

The value of the parameter θ for a full sequence was calculated by converting each θ to a reference temperature and equivalent time. These times were then added and the overall θ found for the sum of the equivalent times and for the reference temperature.

For the data shown, the value of the constant C in the parameter was chosen at 17 for the best fit.

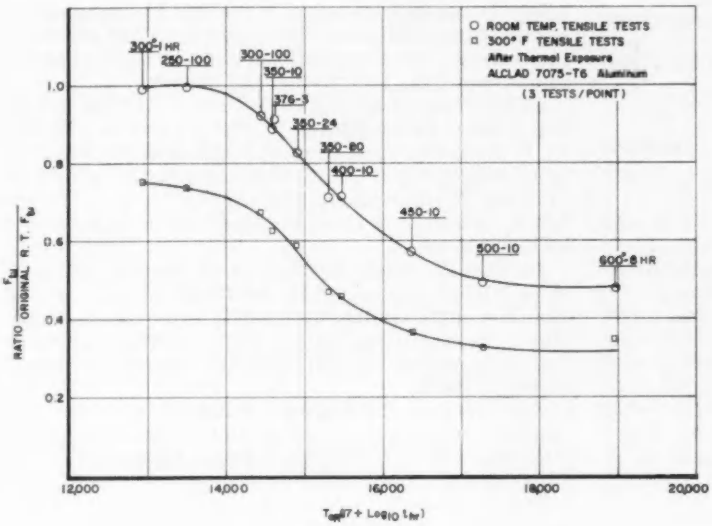

Fig. 1 — Tensile strength of 7075-T6 aluminum for single temperature-time exposures varies smoothly when plotted against the exposure parameter $\theta = T(C + \log t)$.

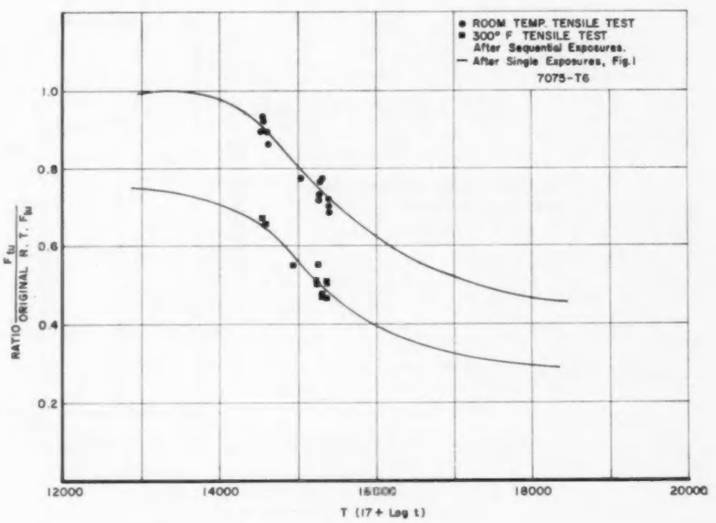

Fig. 2 — Tensile strength after complex exposures compared with strength after single exposures.

This constant was found to vary with different lots of 7075-T6, in one case being 13.5. In some alloys, the best value of C seems to change slightly between the moderate exposure region and the more severe exposure region.

Mechanical strength characteristics

Positive results for the ultimate tensile strength test predict that similar results will be forthcoming for yield tensile strength, compressive yield strength, and strength in shear and bearing. Confirming tests shown in Figs. 3–7 bear this out.

To illustrate the similar characteristics in these curves, vertical (dashed) guide lines have been drawn at key exposure values. There is agreement in the degree of exposure at which strength loss begins and again in the degree of exposure at which the process is essentially complete. Furthermore, the transition curve has the same general shape in all cases, although the range of total strength change does differ.

Further comparison of the various transition curve characteristics can be accomplished by relating the strength at any degree of exposure to the interval between the initial strength and the annealed strength. This approach is illustrated in Fig. 8. Here, the ratio of the remaining strength increment above annealed strength to the total strength range expresses the position of the strength-after-exposure value in the range of possible values. This ratio has been called the strength deterioration factor and is written:

$$D = \left. \frac{F_\theta - F_a}{F_o - F_a} \right|_{T_{\text{test}}}$$

where F_θ is the strength after exposure θ, F_o is the

initial strength, F_a is the annealed strength, and T_{test} is the temperature of testing for all values.

Two reference values of strength are needed to use this relationship, F_o and F_a, and these are needed for each type of strength and for all temperatures under consideration. For 7075-T6, the needed reference values are shown in Fig. 9–13. The F_o values in each case are strength at temperature after no exposure, and the F_1 values are strength at temperature after a reference exposure of $\theta = 16{,}380$ (450 F for 10 hr). Use of F_1 (nearly annealed) values instead of F_a (fully annealed) values is a matter of convenience and avoidance of scatter found in the fully annealed test values; this introduces negative values of the D factor, but does not change the concept.

Comparison of all the 7075-T6 data of Figs. 3–7 on this basis is accomplished in Figs. 14 and 15. In Fig. 14, tension ultimate, tension yield, and compression yield strengths at room temperature 200 F, 300 F, and 400 F after single and multiple exposures are found to correlate closely enough to be represented by a single curve.

In Fig. 15, shear ultimate and bearing ultimate strengths are compared to the single curve from Fig. 14. The same trends of the data are evident as before, although there may be a tendency for the shear strength curve to fall slightly below the other strength data.

The general correlation achieved provides the basis for procedures for prediction of strength of 7075-T6. Given a complex thermal exposure history,

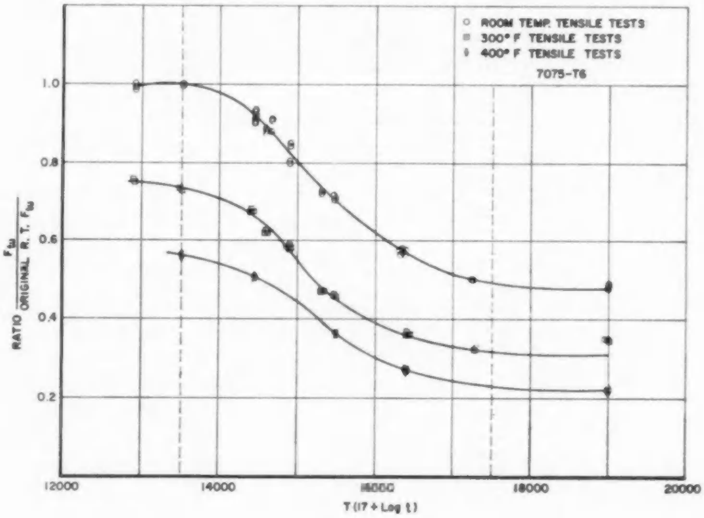

Fig. 3 — Tensile ultimate strength variation with thermal exposure.

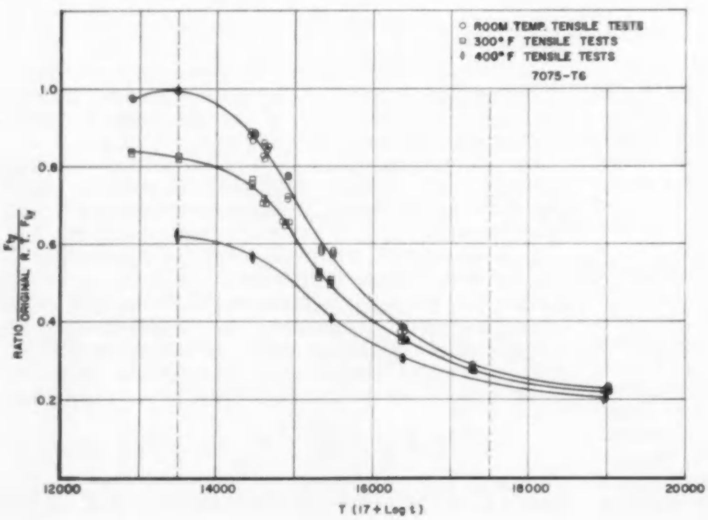

Fig. 4 — Tensile yield strength variation with thermal exposure.

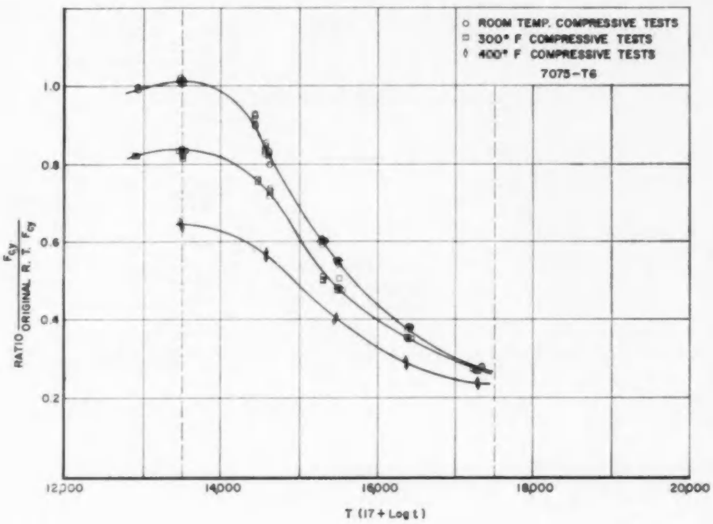

Fig. 5 — Compressive yield strength variation with thermal exposure.

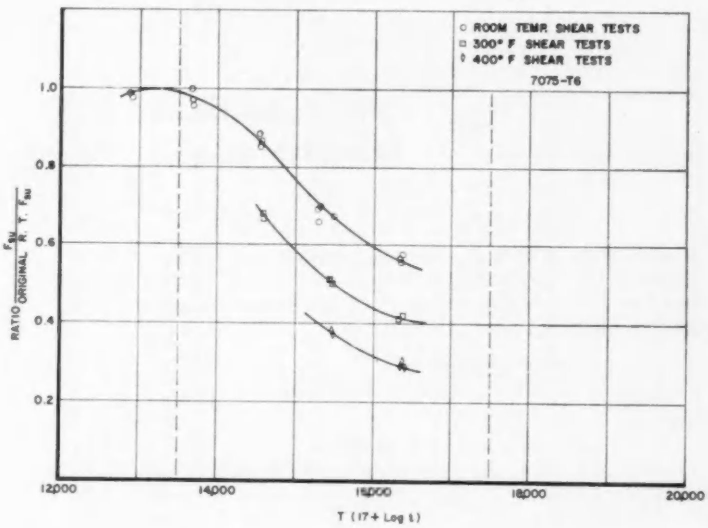

Fig. 6 — Shear ulitmate strength variation with thermal exposure.

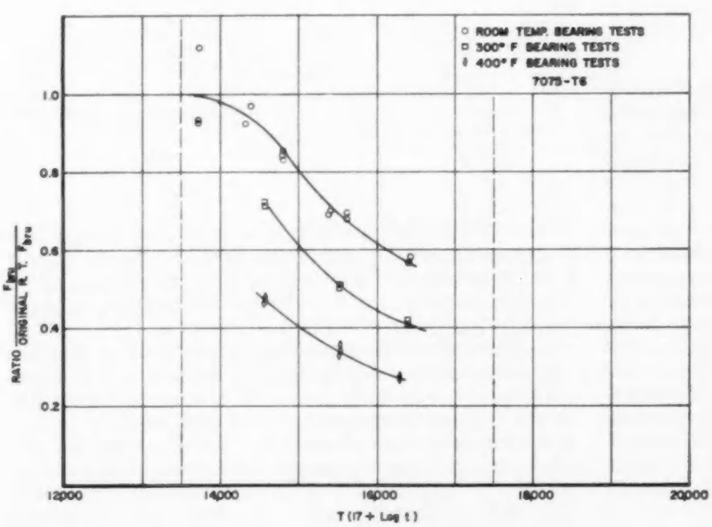

Fig. 7 — Bearing ultimate strength variation with thermal exposure.

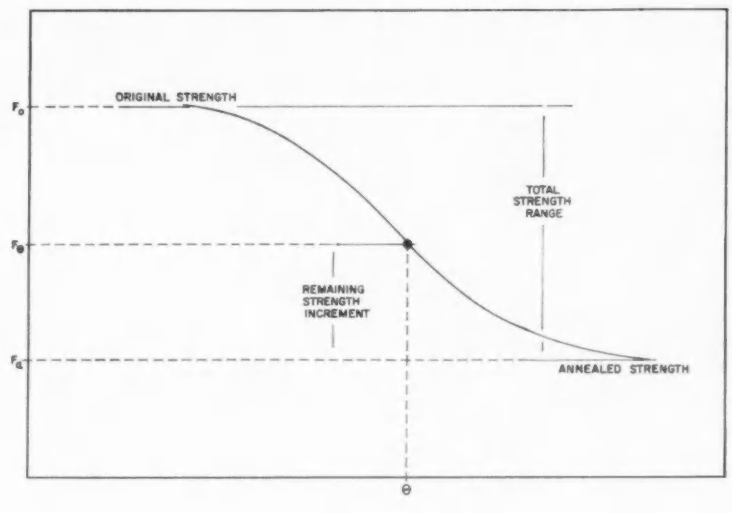

Fig. 8 — Strength after exposure related to total strength range.

High-temperature strength prediction

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the first step is to find the total exposure θ. Next, the strength deterioration factor D is found either from the curve of Fig. 14 or from the empirical expression for this curve:

$$D = \frac{34.80}{(\theta \times 10^{3} - 12.92)^{4} + 28.95} - 0.202$$

where:

$$13{,}000 < \theta < 16{,}380$$

Once D is known, the reference strength values for the type of strength and the temperature desired are chosen from Figs. 9–13. Finally, the desired strength is found from the following relationship:

$$F_\theta = DF_0 + (1 - D)F_1 \Big|_{T_{\text{test}}}$$

This success with 7075-T6 graphically demonstrates the value of both the rate process approach to the problem of material over-aging and the hypothesis that all types of strength symptomatically reflect the same basic change in the material and hence tend to exhibit similarities in characteristics. However, there is no firm basis for expecting all mechanical properties to fall exactly on the same D curve. Ductility for instance, would have to follow an entirely different shape of curve, increasing in some fashion with increasing exposure. Similarly, any mechanical property affected by ductility, such as notched tensile strength or tear resistance, should also be influenced in this direction. Thus, the tendency for the shear data in Fig. 15 to fall slightly below the curve probably represents a real difference in the curve shapes. Tensile yield and ultimate strength values may also follow distinct paths, which differ slightly in Fig. 14.

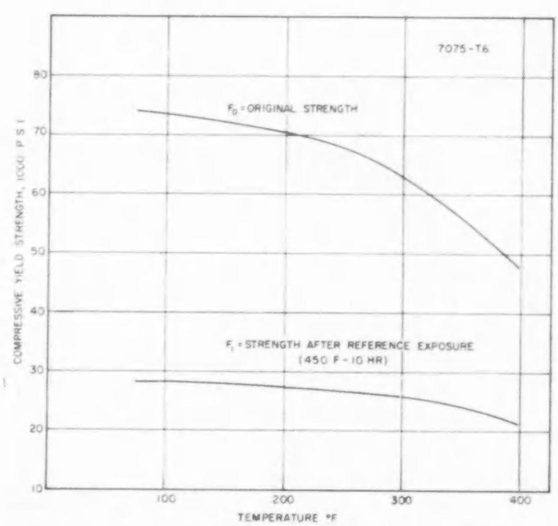

Fig. 11 — Compressive yield strength versus temperature before and after a reference exposure.

Effects of stress during exposure

Exposure-stress sufficient to cause inelastic strain has a detrimental effect on residual strength, studies show. However, for 7075-T6, the effect is small for practical degrees of strain.

In Fig. 16, tensile ultimate, tensile yield, and compressive yield strength, after tensile stressed exposure resulting in 0.2% and 1.0% accumulated inelastic strain, are compared with the unstressed strength deterioration curve from Fig. 14. These curves are used to evaluate strength after stressed exposure similar to that discussed for Fig. 14, as long as the degree of inelastic strain is known. For designs in

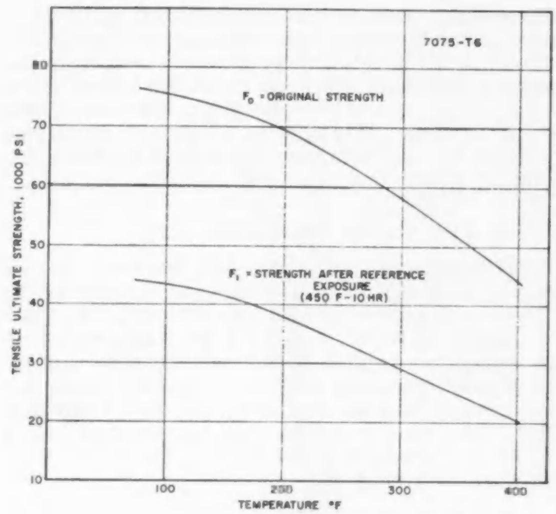

Fig. 9 — Ultimate tensile strength versus temperature before and after a reference exposure.

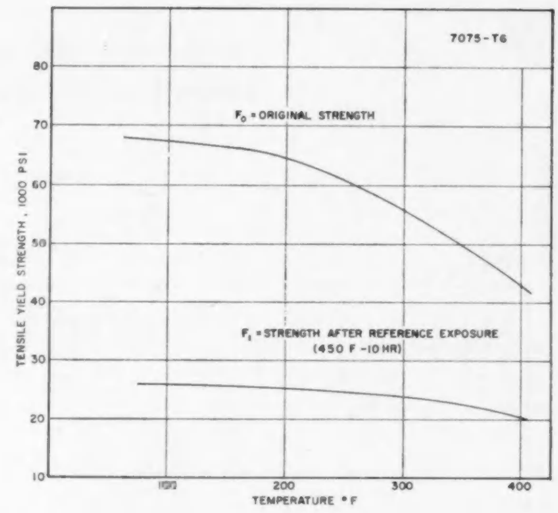

Fig. 10 — Tensile yield strength versus temperature before and after a reference exposure.

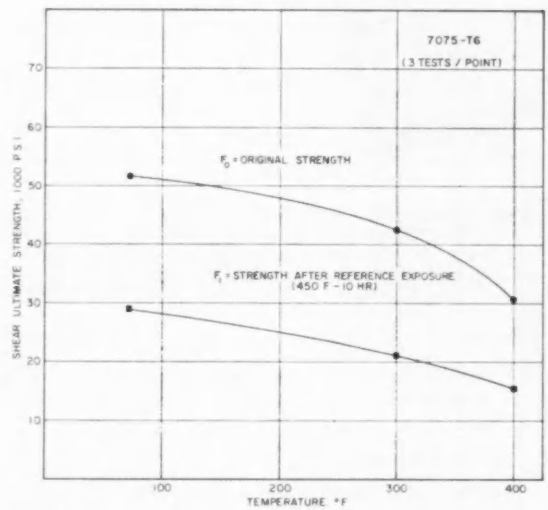

Fig. 12 — Shear ultimate strength versus temperature before and after a reference exposure.

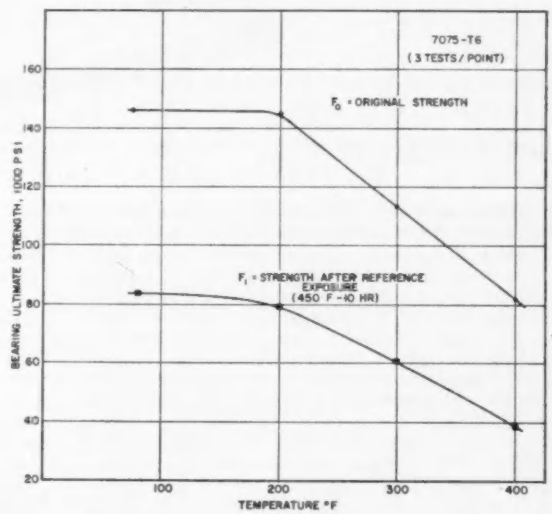

Fig. 13 — Bearing ultimate strength versus temperature before and after a reference exposure.

which strength at the end of the exposure life is critical, the maximum permissible inelastic strain is frequently 0.2%. For such cases, the effect of stress during exposure is small and can be simply accounted for by using the 0.2% exposure curve for all situations, rather than attempting to determine an exact value of accumulated strain between zero and 0.2%.

The stressed exposure curves of Fig. 16 may be considered accurate for cases where the straining either occurs early in the exposure period or is reasonably distributed throughout this period. Check tests have indicated that they are also sufficiently accurate for straining prior to exposure such as might occur during manufacturing, where limited forming may be done in the T-6 condition. Other tests have shown that strain which accumulates very late in the exposure history will have less effect in reducing final strength than strain accumulated early or throughout the exposure.

The largest effect of inelastic strain during exposure occurs at the lowest exposure values. In this region, the Bauschinger effect is evident in that prior straining causes an increase in the yield strength for restressing in the same direction and a decrease in yield strength for the opposite direction

High-temperature strength prediction

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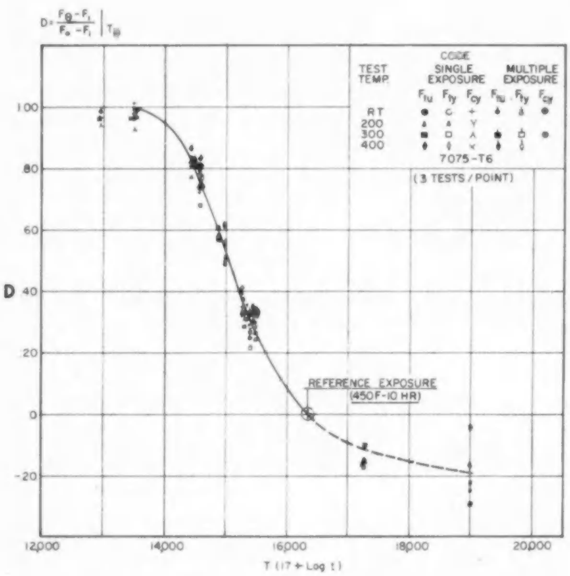


Fig. 14 — Tensile and compressive strength deterioration with thermal exposure for all conditions tested. ($T_\theta = T_{test}$)

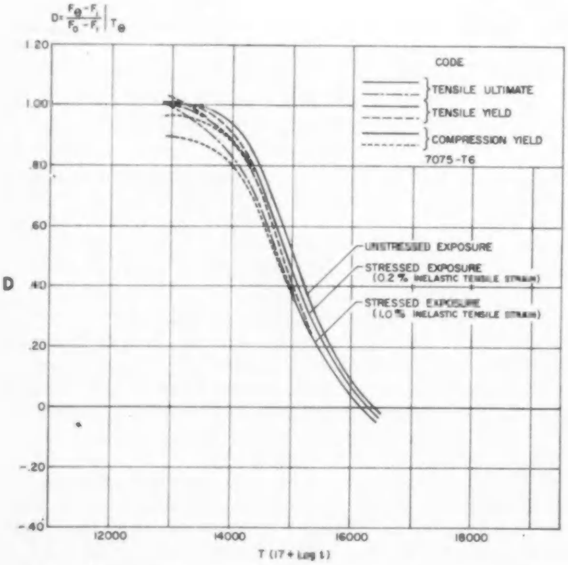


Fig. 16 — Tensile and compressive strength deterioration after (tensile) stressed exposures. ($T_\theta = T_{test}$)

of stressing. This should be borne in mind in using these curves for cases where exposure stresses in one direction may be followed by critical stressing in the opposite direction; for such cases the lowest curve in Fig. 16 (F_{cy} curve) for the given degree of strain should be used. This approach has been confirmed by check tests for compressive stressed exposure followed by tensile testing.

Other applicable materials

This basic approach to strength deterioration has been applied with fair to very good success to the following aluminum alloys: 2024-T3, 2024-T86, 2014-T6, X2020, 2618-T61, and 7079-T6. Agreement on the tempering process in martensitic steel has already been demonstrated, and this approach has also been applied to the age hardening of titanium alloys. At present, three high-temperature alloys are being investigated; PH15-7Mo, 301 Extra Hard Stainless, and Rene 41.

To Order Paper No. 228C . . .
from which material for this article was drawn, see p. 6.

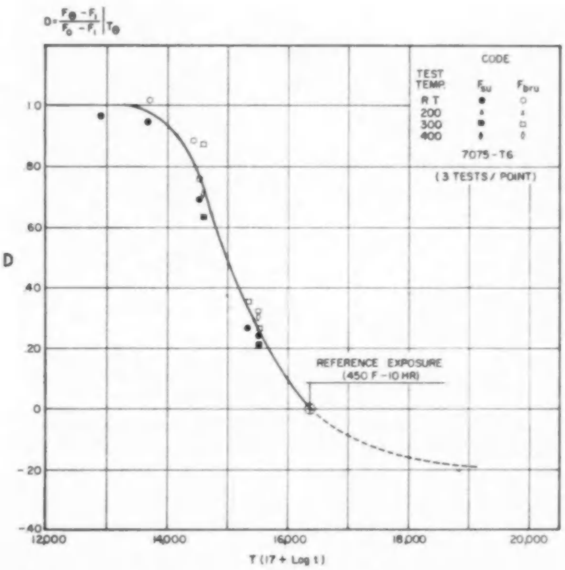
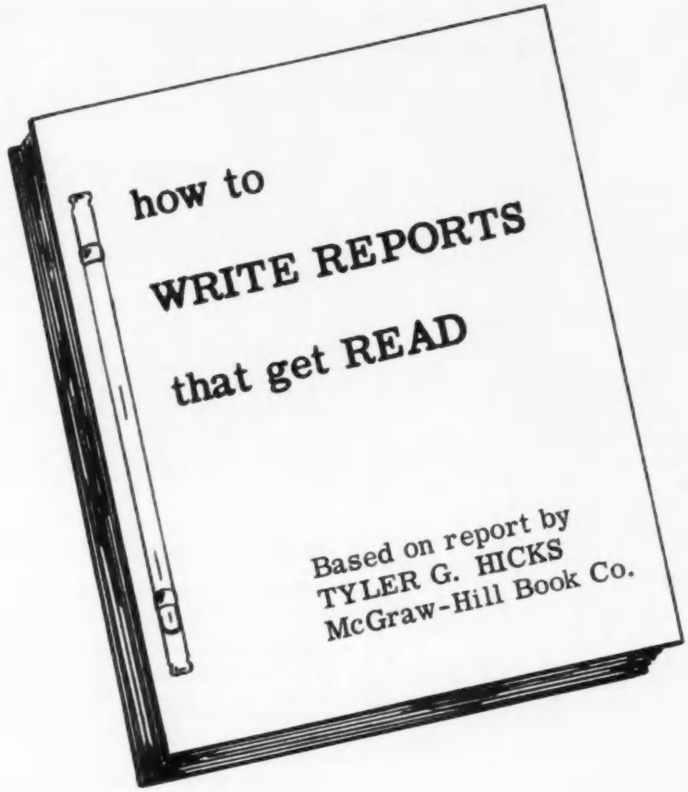


Fig. 15 — Shear and bearing strength deterioration with thermal exposure for all conditions tested. ($T_\theta = T_{test}$)



MOST reports are written to make the reader act. His act is a decision — for or against a proposal.

Study a dozen formal, nonroutine technical reports and you'll see that almost all are concerned with a subject that eventually requires a decision. For instance, see if you can name the decisions that will probably be made on the basis of the following six reports:

(1) Strength Characteristics of Cadmium-Plated Trunk Hinges.

(2) High-Speed Stability of Imported Sports Cars.

(3) Stress Concentration in Machine-Base Fillets.

(4) Performance of Gas Turbines as Pipeline Compressor Drives.

(5) Economics of Diesel Taxicabs in Intercity Service.

(6) Ten Schemes to Reduce Snow-tire Road Noise.

The decisions will probably be:

(1) Cadmium-plated hinges do or do not have enough strength for use in auto trunks.

(2) The high-speed stability of imported sports cars is greater or less than the stability of domestic sports cars.

(3) Stress concentrations will or will not permit use of fillets in machinery bases.

(4) Gas turbines are or are not superior to other prime movers on pipelines.

(5) Diesel taxis are or are not more economical than other cabs in intercity service.

(6) Snow tire noise can be reduced.

Note how far-reaching the decisions based on these reports may be. In (4) several million dollars might be spent for new gas turbines, if they are proved more economical than other prime movers. A large expenditure might also be made if diesel taxis (5) return their investment sooner than other types. Accurate reports help management reach important decisions like these quickly. Inaccurate reports might cause wrong decisions, leading to a loss of profit, prestige, and customer goodwill.

Recent studies show that some engineers shun the profit aspects of their work. These men prefer to create new designs slowly, without a delivery date urging them on. Management aims to early introduction of the new designs created by engineers because these designs promise greater sales, higher profits, less customer dissatisfaction. So the aim of the two — engineering and management — seem opposed. Engineers approach a new design from a scientific base, management from a profit base.

But the aims of engineers and managers really are the same. Engineers seek advancement in their jobs by turning out better designs. These better designs help increase the sales and profits of the firm employing the engineers. The manager uses the work of the engineer to increase the income of the firm. Increased profits help everyone in the firm — from the president to the newest junior engineer. So although engineers and management work from different bases, the aim of their efforts is the same — to turn out better products for a greater number of people.

Once an engineer recognizes where his efforts fit in the structure of his firm he immediately senses that his reports are important. And he sees that clear, sharp reports will aid management men — particularly those who are not too familiar with the technical aspects of a new project. So, today we find many engineers wondering: How can I write reports that will help management but that still retain technical integrity? Is there a way of planning a report to satisfy both needs? The answer is yes. Here it is.

Plan reports for results

We now know that most formal reports are written to make or help the reader decide something. To make your

how to WRITE REPORTS that get READ

. . . continued

reader act you must catch and hold his attention for the time it takes him to read your report. To get the results you want from every report you write:

1. **Choose** the action you want your reader to take.
2. **Select** an approval-getting summary.
3. **Write** the report in terms that will get fast approval.
4. **Arrange** your report for easy reading and use.
5. **Verify** facts, English, presentation.
6. **Deliver** the report so it gets full attention.

Note how different this approach is from the casual "Oh, I'll write a fast report on this for the guys upstairs." Certainly, fast reports are and can be written. But if you want to consistently get results from your technical reports you must take time to plan for "yes" results.

Choose action you want[1]

Use crystal-clear terms for the action or decision you want from your reader. Be certain you state exactly what you think should be done. It is absolutely necessary that you put the action or decision into clear, simple terms. For if you can't put the action or decision into words, how can you expect the reader to guess what you mean? Here are examples of clear decisions your reader can either approve or disapprove:

Cadmium-plated trunk hinges should be used in new cars.

Machine bases should be cast with fillets.

Gas turbines should be adopted as prime movers.

Diesel taxis should be purchased for intercity use.

Note that all these proposed decisions are stated in the form of recommendations. Why? Because it is easier for you to state the decision you want in the form of a recommendation than in any other way. A recommendation is direct, concise — it says either we should or we should not. Also, a clearly stated recommendation is easier for your supervisor to approve than a long, involved statement of what is proposed, why it is proposed, what it will do, etc.

TYLER HICKS has examined hundreds of engineering reports . . . and traced the results they get and don't get.

Much of what he has learned from these years of analysis is focused in his book titled, "Technical Report Writing" . . . and, in its direct application to automotive engineers, in this article.

Hicks is editor of engineering and industrial books at McGraw-Hill Book Co.; was formerly a technical editor of *Power*; and has taught mechanical engineering at Cooper Union in New York.

But what about a report that does not have a recommendation as its conclusion? How should we indicate the action or decision the reader is to make? Use a similar scheme — state your thoughts in clear, strong terms. Thus, your report might say:

Downdraft carburetors are the most economical for the xyz engine.

Sodium-cooled valves have an average life of 30,000 hr.

Flatter torque curves are desirable in marine gasoline engines.

Hydraulic reverse-reduction gears are used in 70% of the engines.

Each of these statements clearly specifies the main thought you wish the reader to obtain from your report. There is no doubt in his mind as to what action he should take, or decision he should reach. Of course, the final decision may hinge on other factors that are not covered by your report. But you have expressed your findings clearly enough so they can be easily considered in the final decision or action. Once you've done this you are ready to select an approval-getting summary.

An approval-getting summary[1]

Each of us has his specialty in the business or technical world. We may be design engineers, research scientists, department managers, vice-presidents, presidents, or even members of the board of directors. In any position we look at the world through the eyes of our specialty.[1] Thus a member of the board of directors is primarily concerned with the business and financial aspects of a report (see illustration). In preparing a report for a board member or the president of your firm you would emphasize the financial and business aspects of the data — not the technical aspects. So instead of saying:

Downdraft carburetors are the most economical for xyz engines

you might write:

Downdraft carburetors save $12 per xyz engine and are as reliable as updraft carburetors.

Reading this, the business-minded board member or president (both of whom are management personnel) immediately translates your findings into savings or profits for the company. Were you writing this report for the production department you might write:

Downdraft carburetors save $3 per xyz engine in installation cost; reliability is as good as updraft carburetors.

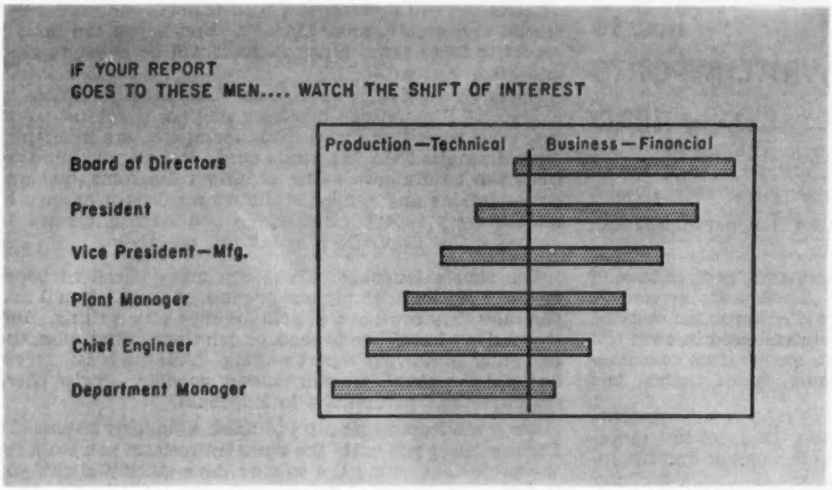
The chief engineer looks at the world from an engineering viewpoint. To him you might write:

Downdraft carburetors are as efficient (30%) as updraft; they reduce the cost of xyz engines 1.5%.

Up to now we've dealt with information that could be presented in one or two sentences. Actually, few reports can be summarized in a single sentence. For most reports you'll need a paragraph — say up to 100 words. This is your key to an approval-getting presentation: *Start your report with the summary. Present the important findings in a single paragraph of about 100 words, or less.*

Note that this summary paragraph is not the place to tell your reader that you tested 273 different careburetors on 191 engines from the Arctic Circle to the Sahara Desert. *The fact that you were assigned to write the report shows that your supervisors have faith in your technical abilities.* They assume you tested enough configurations in a variety of atmospheres to reach valid conclusions. In your sum-

[1] "Writing Reports that Get Action," W. A. Ayres. "Factory Management and Maintenance," September, 1955.



A technical report should be written to interest the particular person or group at which it is aimed.

mary try to answer the questions the reader is likely to ask. These are:

What's in it for us? (Profits; lower costs; time and labor savings; greater reliability; less maintenance, etc.)

What's the cost, weight, size, performance efficiency?

What are the advantages and disadvantages?

How will this help our business? (Competitive aspects; field problems; new-product aspects, etc.)

There are few reports in which your summary must answer every one of these questions. But be sure to supply an accurate answer to every question you think *might* be raised. Direct the answers at the major interests of the person to whom you are submitting the report. Thus, a report to management will have a summary different from the same report prepared for the chief engineer of a firm.

Here's an example for management:

Downdraft carburetors will save $12 per xyz engine and are as reliable as updraft carburetors. The total saving of $12 is made up of these cost reductions: manufacturing $5; installation $3; controls $4. A change from the existing carburetor to the downdraft type could be made without interfering with present production. Three suppliers are available who will guarantee delivery of 1000 carburetors per month. Quality is equal to that of carburetors we now use. Current production rate of the xyz engine is 900 units per month.

For engineers the summary might be worded like this:

Downdraft carburetors are the most economical type for the xyz engine. They show an efficiency of 30%, the same as the updraft type now being used. Cost of the engine is reduced 1.5% when using the downdraft carburetor. Reliability equals that of present carburetors; manufacturing, installation, and controls are simpler. Tests show mixture ratios are accurately maintained through the full normal load and overload ranges of the engine.

Ask a hundred engineers or executives what their biggest daily chore is and they'll almost all give the same answer — reading. Engineers and managers in key industries say they are flooded with reports from inside and outside their firms. Few of these men have the time to read any but the most important reports reaching their desks. But almost every astute engineer and manager is glad to read a well-prepared summary of a report. Why? Because in a few moments the intelligent engineer or manager can get all he needs to know about a given subject from an accurate report summary. The summary gives him enough information to make a decision. And this, as we saw earlier, is the ultimate aim of almost every report written today. Once you've prepared an approval-getting summary, you're ready to write the report.

Write for fast approval[1]

The way you write your report can mean the difference between approval and rejection of your proposal. Since refusal of a proposal means the time and energy spent on the report were wasted, make every effort to write the best report you can. You *can* write better reports. How? Follow these guides:

a. **Aim** at your probable reader.

b. **Summarize** data to speed reading.

c. **Tabulate contents** for easy data location.

d. **Tab main sections** for quick use.

e. **Choose** effective illustrations.

f. **Use** simple language.

Aim at your probable reader: Earlier, we saw how you can tailor the intent and summary for a given class of readers — usually engineers or managers. You can't drop your good intentions once the summary is written. Though only 10% of your readers may go beyond the summary, you still have to catch their attention and make them act. To do this you must write the report so it covers their main fields of interest.

When you know the report will be read primarily by engineers, give engineering data the main emphasis. For managers, emphasize the business aspects If the report is to be read by both groups, break it into two sections. Label one "engineering data" and the other "management or business data." Then the reader can find what interests him without wading through pages of data that are of no value to him.

Use an outline to guide your writing. List the topics you wish to cover. Start with the important items and add the topics in an order of decreasing importance. Never try to write a report without first making an outline. You may get the report written without an outline, but it will take more time and be more painful to you. Once you've listed all the topics, group them into the section outline for a typical report, given below.

Summarize data to speed reading: Use a summary paragraph to begin each section of the report. Thus, in a section on test procedures, summarize the procedures used be-

how to WRITE REPORTS that get READ

. . . continued

fore describing any in detail. A typical summary paragraph would be:

> Fourteen tests were run to determine the performance of the ABC downdraft carburetor. These tests were made on seven gasoline engines with displacements ranging from 150 cu in. to 600 cu in. Data collected in each test include brake horsepower output, specific fuel consumption, oil temperature and pressure, spark timing, and friction horsepower.

Following this summary paragraph the exact test procedures are described, along with the equipment and instruments used. Findings of each test are tabulated for later reference or verification.

A series of summary paragraphs of this type allows interested readers to make a quick review of what you did. Should the reader want more information he can read the entire presentation in any section important to him.

Tabulate contents[1]: List every major section of the report. Give the page number beginning each section so the reader can see at a glance how many pages are devoted to a given topic. Never slight the table of contents. Sometimes your report will be evaluated by a reading of only two items in it — the summary and the table of contents. This may seem unfair but in the press of business activities people develop various schemes to save time. And the two-item reader is common in business today.

Tab sections for quick use: Index tabs[1] are another aid to faster use of your report. Use a tab for each important section. Give the tab the same title as the section it indicates. For if you use different titles for the tabs the reader will be confused. Tabs are usually more effective than colored pages for separating report sections because a tab is easier to locate. Remember — you are competing for the reader's time. Every minute you save him is that much more time he can spend reading the report.

Choose effective illustrations. We all know the proverb about a picture and a thousand words. Like all proverbs this one is not always true — particularly for reports. For a poor illustration or one not appropriate for the data in the report can waste time and space. As examples, illustrations of standard test equipment like electric dynamometers, prony brakes, CFR engines, flowmeters, etc., waste space in a report, unless they show new hookups or arrangements. So, omit this type of illustration. In its place use illustrations showing equipment or modifications to equipment the reader is not familiar with.

Don't think the only illustrations you can use in a report must be technical. Others — both semitechnical and nontechnical — can be used, *if they are pertinent and convey useful information to the reader*. Every illustration must be tested for its usefulness. Ask yourself: Will my typical reader obtain any helpful information from this illustration? If the answer is yes, include the illustration.

Every illustration need not be a drawing, photograph, or graph. You can use newspaper or magazine clippings, sample pages from catalogs or instruction manuals, business forms, pertinent letters from important persons or companies, memorandums, excerpts from other reports, etc. So long as the illustration is pertinent to the report and helps inform the reader, use it. Of course, you should not load a report with so many illustrations that the reader finds it impossible to wade through them. You must use judgment. Though there are no rules stating how many illustrations can be used in a report, few technical reports should ever require more than 50. And if you can limit a report to 25 or fewer illustrations it will be easier to read and use.

Never overlook the chance to use dramatic illustrations in a report. Illustrations depicting motion, people at work, machines performing interesting tasks, etc., are inherently more dramatic than flat, static ones. Not all your illustrations can be dramatic — use of circuit diagrams, machine cross-sections and similar static drawings is necessary in almost every report. But where you have a chance to choose a few dramatic illustrations, use them.

Use simple language: There are many excellent books on clear writing.[2] Study one or more of these. You'll find the rules they contain will help improve your writing. But if you don't have time to read, or can't remember rules, use one guide in all your report writing. This guide is: *Choose and use the simple word whenever possible. Prefer short sentences and paragraphs to long ones.*

Once you start writing, try to finish as quickly as possible. For the faster you write the more enthusiasm you build for the project. Getting the writing done quickly allows you less time to feel sorry for yourself, or to daydream about what you'd rather be doing than writing a report. Remember: The only way to get words on paper is by writing — no amount of thinking and talking will do the writing for you. So, think as you write and use your writing as a substitute for talking. You'll be amazed at how much your writing output rises.

Forget the intricate rules of grammar when you write. Get the technical facts on paper. Later, after all the data are in rough form you can revise and improve your grammar.

Arrange report for easy use[1]

Long industrial experience shows that one desirable arrangement for the sections of a report is: title, table of contents, conclusions, preface, body, appendices. Let's take a quick look at each.

Title: Pick a short, accurate title for your report. Be sure the title is not misleading — executives will feel you wasted their time if the title promises one thing but the report gives another.

The title will appear in three places in the report — on the binding or cover, on the title page, and in the letter of transmittal. Typical report titles used in industry today tend to be long. A long title can be justified if it is accurate. But, where possible, use as short a title as possible. Try to inject interest into the title by including the positive aspects of your findings. Thus you might write:

Automotive Air Conditioners
Increase Driving Safety

instead of:

Report on
Studies of Automotive Air Conditioners

Certainly the first title is more interesting than the second.

Center the title on the report binding and title page. Use a subdued color for the report binding. Unless your firm

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[2] "The Way to Write," R. Flesch and A. H. Lass. McGraw-Hill, N.Y., 1955.
"Technique of Clear Writing," R. Gunning.. McGraw-Hill, N.Y., 1955.
"Guide to Technical Writing," W. G. Crouch and R. L. Zetler. Ronald Press, N.Y., 1954.
"Successful Technical Writing," T. G. Hicks. McGraw-Hill, N.Y., 1959.
"Clear Writing for Easy Reading," N. G. Shidle. McGraw-Hill, N.Y., 1951.

has rules concerning the color to be used, white is the best color for the title page, regardless of what your secretary might think.

Table of Contents: Use the section headings as entries for the table of contents. If you err in preparing the contents, err on the side of more entries than less. For the user of your report is generally in a hurry. The more information you put into the contents, the better will he appreciate the report.

Be sure the entries in the contents are descriptive. Many people will "read" your report by giving the table of contents a quick glance. Misleading entries will leave them with an incorrect impression. Thus:

I. Automobile sales in Idaho:
 a. Western.
 b. Southern.
 c. Eastern.

tells the report user very little. Change this kind of an entry to:

I. Number of automotive sales in Idaho:
 a. Western sales decreased in last two years.
 b. Southern sales up 20% in two years.
 c. Eastern sales hold steady.

The second approach to the table of contents tells the reader what to expect in the report.

Conclusions: Make them as short, accurate, and pertinent, as summaries, which were discussed earlier.

Preface: Not all reports need a preface. Reports that will be read by a large group of people, or the general public, benefit from a preface. For here the report author can *briefly* tell his readers the subject of the report, why the report was written, investigation methods used, and from whom help was received. The tone of the preface is less formal than the report itself.

If you wish to have an authority or other well-known person or firm say a few words concerning why the report is needed, put this information into a foreword. In many reports the foreword precedes the preface and tells how useful the report is.

Body: The body contains the bulk of relevant data on the subject of your report. Typical information included in the body is authorization, purpose and object, scope and method, limitations, definition of terms, apparatus and materials, outline of procedures, acknowledgments, bibliography and references. For the reader the body of the report is where the problem is defined, analyzed, and solved. It is here that he expects to find all the analytical data related to the problem. But in writing the body of your report, avoid inclusion of lengthy tables, calculations, and similar data.

Appendices: Here is where long tables, mathematical proofs, calculations, progress diagrams, analytical charts, etc., belong. This leaves the reader free to ignore these data and concentrate on the conclusions and body of the report. Should the reader want to verify an equation he can easily do so by turning to the appendix. But it is not necessary for him to read through every deviation, calculation, or table to understand the report.

Check the report

Once the report is written, review it completely before having the final draft typed. Verify every fact by comparing the data in the report with the original information. Check the English. If your grammar is poor, admit this to yourself. Get someone in your office to read the report and mark the errors for you. By making a few discreet inquiries you can easily find one or more people near you who are well qualified to criticize the grammatical aspects of the report. Follow their recommendations; the report will usually be a better document if you do.

Examine the report arrangement. See that it follows the general outline given above. If necessary, you can change the section sequence. But be sure there are good reasons for doing so. Of course, if your firm has a standard sequence for reports, use it. For if you use a nonstandard sequence, the report has little chance of acceptance.

When reviewing the report be as critical as possible. Every improvement you make before submitting a report is one less item that can be criticized by the person or persons reading the report. Remember — you cannot sit beside the reader and explain yourself as the report is read. The report must stand on its own. As such, it should be the best you can produce at the time. Never forget that it is far easier to criticize a piece of writing than to produce it. So we find that the world is full of critics but short of good writers — particularly of industrial reports.

Deliver the report[1]

Some engineers and scientists spend months on a report and toss the finished job into the interoffice mail for delivery. This is just about the worst way possible to deliver an important document. If a report is worth spending many hours of your life on, it also rates proper delivery.

Check to see how the report originated. If one of your supervisors requested it verbally or by memo, make an appointment to see him. Do not leave the report with the man's assistant, or his secretary. Wait until the supervisor has time to see you. When he does, bring him one copy of your report. Remind him of his request for the report to be written. Then briefly summarize your findings. Since the supervisor is probably a busy man, confine your summary to the important findings. Don't try to explain every step you took in preparing the report. The supervisor isn't interested — what he wants are results.

Be ready to answer questions about the report. Give a direct *yes* or *no* answer to every question. Avoid detailed answers, unless they are requested. Try to keep in mind the location of various items in the report. If the supervisor asks "Did you test these carburetors under freezing conditions?" Answer "Yes, that's covered somewhere around p. 20."

Before leaving the supervisor tell him you'd like to know what he thinks of the report, after he has read it. Most supervisors will honor such a request, sending you a note, or calling you on the phone to give their opinion. If the opinion is complimentary, as it almost certainly will be if you planned and wrote the report well, you will be encouraged. And the supervisor will remember you.

If you're ever called at a later date to discuss the report, take time to read the report completely *before* the discussion. For if a month or more passes before you're called for the discussion, you'll have forgotten many of the details of the report. And nothing is so unconvincing as an engineer or scientists who can't adequately defend his own report.

Good reports pay off

You never lose when you write a good technical report. For an effective report can mean the difference between approval and rejection of a project. One good report can be the key to a better personal future; a series of effective reports can put you into an important managerial position.

Effective technical reports can lead to other writing projects — technical articles, society papers, even technical books. In these forms of technical writing your name may become the byword of your field.

But if these promises mean nothing, then the words of Phil Swain, former editor of *Power* magazine, and outstanding exponent of clear writing may convince you. He said "Other things being equal, the engineer who writes well will earn $20,000 to $50,000 more during his career than the engineer who can't or won't bother to write well."

How Engine Operating End-Gas Pressure-Temperature

Based on paper by

M. E. Gluckstein and C. Walcutt

Ethyl Corp.

ENGINE END-GAS temperature-pressure histories were studied at knock-limited compression ratio for different engine operating conditions. It was found that increasing engine speed, advancing ignition timing, or increasing fuel-air ratio from lean to stoichiometric will cause higher final end-gas temperature. Large changes in inlet-mixture temperature had no effect on final end-gas temperature.

End-gas temperatures and pressures were measured in a variable-compression ratio, split-head CFR engine using the speed-of-sound and balanced-pressure technique.

Engine variables investigated

Engine variables investigated were inlet-mixture

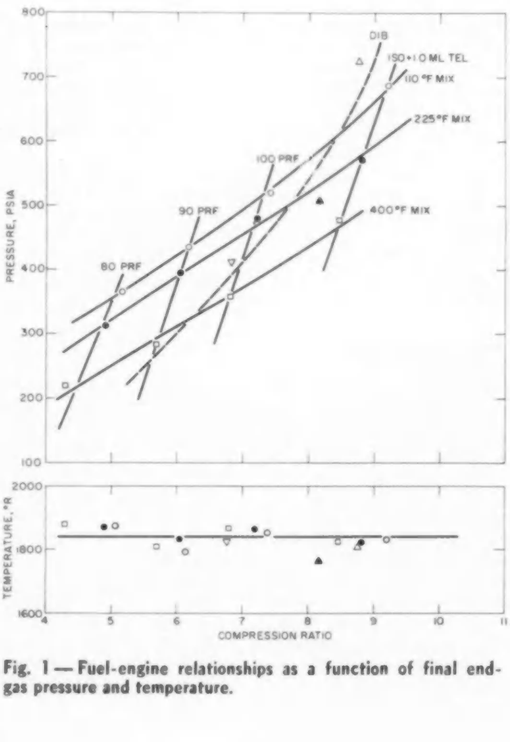

Fig. 1 — Fuel-engine relationships as a function of final end-gas pressure and temperature.

temperatures of 110–400 F; ignition timing of 5–25 deg btc; engine speeds of 608 and 1227 rpm; and fuel-air ratios ranging from 0.056 to 0.084.

Inlet-mixture temperature

For the study of inlet-mixture temperature, the fuels used were primary reference fuels, isooctane plus 1.0 ml tel, and diisobutylene. The engine was adjusted for knock-limited compression ratios at inlet-mixture temperatures of 110–400 F for each fuel. The other engine operating conditions were 15 deg btc ignition, 608 rpm, and 0.070 fuel-air ratio (F/A).

Effects of engine and fuel variables on final end-gas temperature and pressure are shown on Fig. 1. In this figure, compression ratio is plotted against final pressure in the upper part and against final end-gas temperature in the lower part. Separate lines pass through the data points for all primary fuels at each inlet-mixture temperature. Similarly, separate lines pass through the data points for each fuel at different inlet-mixture temperatures. The pressure data lines are separate and distinct, but all lines for the temperature data merge into one within the experimental variability. In other words, at knock-limited compression conditions, all fuels knock at the same final end-gas temperature over a wide range of inlet-mixture temperatures. The major distinction between the fuels and engine operating conditions lies in the pressure values.

The relative constancy of temperature is only a matter of circumstances relating to engine operation and fuel behavior characteristics. Fuel behavior is discussed in a later section dealing with precombustion.

Engine operation relates to the manner in which knockratings are made in a variable-compression-ratio engine. If the inlet mixture is heated, the engine will knock more. To reduce knock, the compression ratio is lowered. This means that the higher inlet mixture temperature, and its effect in producing a higher final end-gas temperature, is compensated by a reduction in compression ratio. The pressure, on the other hand, must decrease due to the decrease in compression ratio.

As an example of the constancy of temperature, the pressure-crankangle and temperature-crankangle relationships for 75-octane number primary reference fuel (prf) and isooctane + 1.0 ml tel at low (nominal 120 F) and high (400 F) inlet-mixture temperatures are shown on Fig. 2.

Conditions Affect Histories

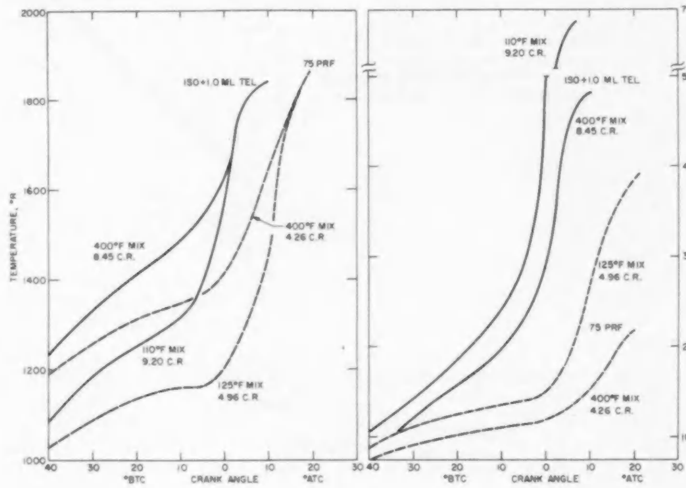

Fig. 2 — Effect of octane number of primary reference fuels on end-gas temperature and pressure (608 rpm, 0.070 F/A).

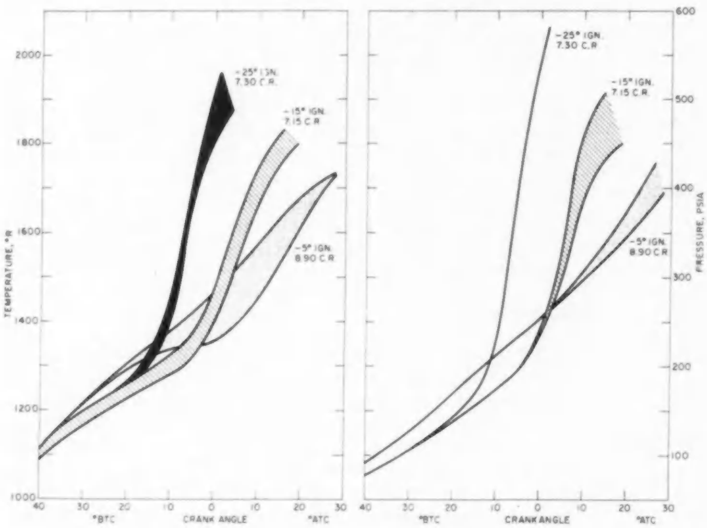

Fig. 3 — Effect of ignition timing on end-gas temperature and pressure.

Before top center, the pressures and temperatures for the 75 prf are considerably lower for a given inlet-mixture temperature than for isooctane + 1.0 ml tel. This is to be expected, since the knock-limited compression ratio for the 75 prf is very much lower.

After top center, the gas temperature for the lower octane fuel at both inlet-mixture conditions continues to rise until the final end-gas temperature before knock is about the same for both fuels.

It is not that the rate of temperature rise is appreciably greater with 75 prf but that the temperature rise with the higher octane fuel was terminated by knock about 10 deg sooner in the engine cycle. A point of some interest is that, although the final end-gas temperatures are about the same for both

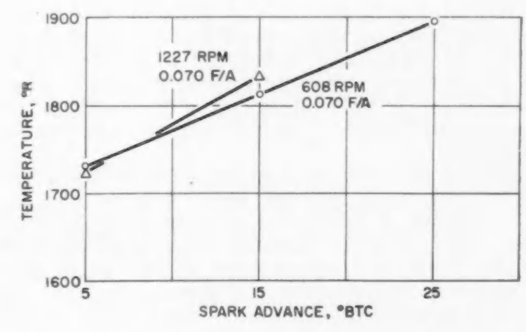

Fig. 4 — Effect of ignition timing on final end-gas temperature (isooctane).

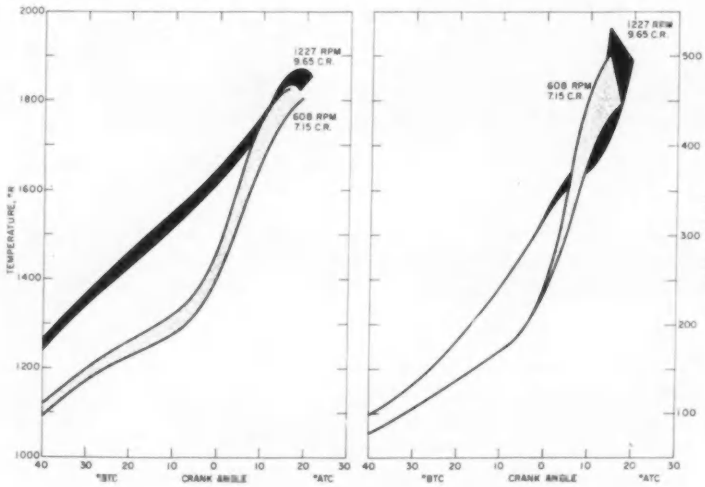

Fig. 5 — Effect of engine speed on end-gas temperature and pressure.

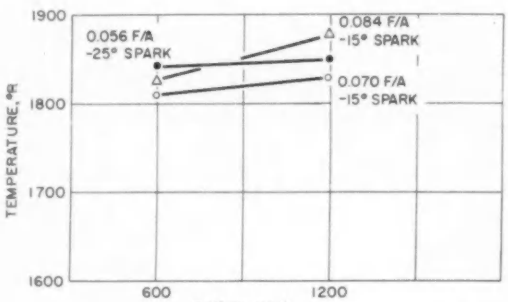

Fig. 6 — Effect of engine speed on final end-gas temperature (isooctane).

End-Gas Pressure-Temperature Histories

. . . continued

fuels at both inlet-mixture temperatures, the maximum difference is 550 R at 5 deg btc.

Turning to the pressure-crankangle data, the difference in pressure between these two fuels at a given inlet-mixture temperature increases throughout the engine cycle. At the time of knock, the maximum pressure difference amounts to 500 psi, in contrast to a negligible difference in the final end-gas temperature. The differences in pressure are in accord with the differences one would expect from the differences in knock-limited compression ratios required.

Based on the premise that pressure, temperature, and ignition delay are the important factors controlling knock, a higher inlet-mixture temperature must require a longer ignition delay since the final cylinder pressure is lower with no change in final end-gas temperature. This is reasonable since the rate of change in pressure and temperature tend to decrease with increased inlet-mixture temperature.

Since knock was held constant in this study, the only requirement on the delay is that it be of sufficient duration to allow all but an end fraction of the fuel to be consumed by normal combustion before autoignition occurs. Too long or too short a delay would mean either too little or too much knock. For each operating condition, the pressure-temperature history must be adjusted by varying compression ratio to produce the correct delay, in order to maintain a constant knock intensity.

Fig. 1 shows that, in the case of diisobutylene (dib), a much larger reduction in pressure is required to maintain the same knock intensity when inlet-mixture temperature is raised from 110 to 400 F. This is also reflected by a much larger change in compression ratio for dib. It is this greater required change in compression ratio for dib that causes its octane value to change from 92 to 104 in this engine when the inlet-mixture temperature is lowered from 400 to 110 F.

Spark advance

To study the effect of spark advance, three fuels were used — isooctane and S and H fuels blended to 100 octane number. S stands for sensitive (ternary mixtures of n-heptane, isooctane, and diisobutylene) fuels and H or HOT fuels are tenary mixtures of n-heptane, isooctane, and toluene fuels. The pressure-crankangle and temperature-crankangle relationships at spark advances of 5, 15, and 25 deg btc are shown on Fig. 3 for 608 rpm, 150 F inlet-mixture temperature, and 0.070 fuel-air ratio.

Curves for the three fuels are nearly identical, as shown by the shaded bands which encompass the corresponding data for the three fuels. The lowest compression ratio and the lowest pressure and temperature before top center occur with the 15 deg btc spark advance. After top center, however, the pressures and temperatures at any given crankangle become higher as the spark is advanced from 5 deg btc.

The effect of spark advance on final end-gas temperatures is shown in Fig. 4 at both 608 and 1227 rpm. A nearly linear relationship exists between increase in final end-gas temperature and spark advance. A 20-deg increase in spark advance raises the final end-gas temperature by 180 R.

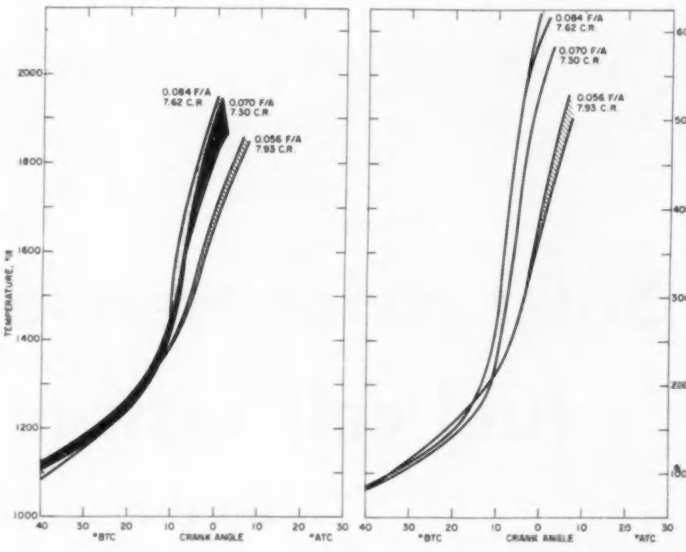

Fig. 7 — Effect of fuel-air ratio on end-gas temperature and pressure (isooctane, 100S, 100H).

Engine speed

The same three fuels—isooctane and S and H fuels blended to 100 octane number — were used to study the effects of engine speed. The pressure-crankangle and temperature-crankangle relationships at engine speeds of 608 and 1227 rpm are shown in Fig. 5 for 15 deg btc ignition timing, 150 F inlet-mixture temperature, and 0.070 fuel-air ratio.

Again, the curves for the three fuels are practically identical as shown by the shaded bands which include the corresponding data points. A greater knock-limited compression ratio is required at 1227 rpm, and this is reflected in appreciably greater pressure and temperatures before top center. Later in the cycle, the pressures and temperatures become more nearly equal. If time is measured on a crankangle basis, the rates of pressure and temperature rise following ignition are greater at the lowest speed — particularly the rate of pressure rise.

The effect of engine speed on final end-gas temperatures for all engine operating conditions investigated is shown in Fig. 6. It is apparent that, for the small range investigated, engine speed has a negligible effect on final end-gas pressures and temperatures.

Fuel-air ratio

The effect of fuel-air ratio was investigated using the same three fuels. The pressure-crankangle and temperature-crankangle relationships at fuel-air ratios of 0.056, 0.070, and 0.084 are shown in Fig. 7 for 608 rpm, 25 deg btc ignition timing, and 150 F inlet-mixture temperature.

Lowest knock-limited compression ratio occurred at 0.070 fuel-air ratio, and resulted in lower pressures and temperatures before top center. After top center, however, the pressures and temperatures for the richer mixtures rose more rapidly. This is undoubtedly caused by the greater energy available in the richer mixtures.

Increasing the fuel-air ratio appreciably above the chemically correct requirements caused a negligible increase in pressure and temperature. This is further indicated by Fig. 8 which shows the effect of fuel-air ratio on final end-gas temperatures at different speeds and spark advances. As shown, changing the fuel-air ratio from 0.056 to 0.070 can cause the final end-gas temperature to increase by about 150 R. Further enrichment has little additional effect.

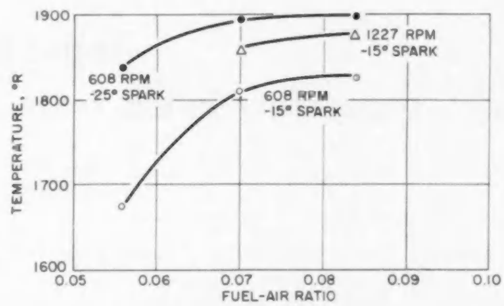

Fig. 8 — Effect of fuel-air ratio on final end-gas temperature (isooctane).

The reason for the changes in final end-gas pressures and temperatures at knock-limited compression ratio for changes in engine operating conditions can be understood on the basis of the factors controlling ignition delay. In general, a change in engine operation which causes a higher rate of pressure or temperature rise just before knock allows the final end-gas pressure or temperature, or both, to reach higher values.

This is in agreement with the basic ignition-delay concept; namely, that a higher pressure or temperature must result in a shorter delay for knock. When the rates of pressure and temperature rise are higher, the time to reach the pressure and temperature obtained at a lower rate must be shorter. If the engine was knock-limited at the lower rates, then the fuel could knock at the higher rate for the same final pressure and temperature as at the lower rate, since the elapsed time would be too short. In other words, the pressure and temperature at the higher rate must continue to rise until the proper combination of pressure, temperature, and ignition delay is obtained for knock to occur.

To Order Paper No. 201F . . .
from which material for this article was drawn, see p. 6.

Solar regenerative fuel cell system on the way

Regeneration and fuel cell phases work separately in lab tests. Next step is to combine them.

Based on paper by

E. Findl, W. B. Lee, J. D. Margerum, and W. E. McKee

Sundstrand Corp.

SOLAR REGENERATIVE FUEL CELL SYSTEM — based on photochemical decomposition of nitrosyl chloride — is being developed successfully at Sundstrand.

This system combines chlorine and nitric oxide in the fuel cell to produce electricity. Nitrosyl chloride formed in the process is subsequently photodissociated into chlorine and nitric oxide. These fuels can be stored for future use or fed back into the fuel cell.

Experimental results to date have been good. . . . Significant amounts of nitrosyl chloride photolysis products have been produced and separated; and a fuel cell which can operate on these products has been demonstrated. Next step is to combine the solar regeneration and fuel cell phases into a single unit.

Choice of system

Studies showed that requirements for fuel cell operation and photochemical regeneration are quite severe and somewhat contradictory.

The regeneration process appeared to be the limiting factor. So emphasis was placed on choosing a promising photochemical reaction which might be adapted to a fuel cell system.

Photochemical Reaction — Of all the photochemical reactions considered, the decomposition of nitrosyl chloride seemed to be the best. A quantum yield of two at wavelengths extending from ultraviolet to about 6370 A had been reported. (This means that for every quantum absorbed, two molecules of nitric oxide are produced.) This spectral band contains about 43% of the total solar energy and about 75% of the photochemically active light (below 8000 A).

The quantum yield of two is obtained through a two-step free radical process, which results in an increase in free energy, ΔF, of 4.9 Kcal/mole. The two products have thus stored part of the energy absorbed from the photon.

Fuel Cell Reaction — The photolysis products give up this energy as electrical energy if they are recombined in the reverse reaction by an electrode reaction in a fuel cell:

$\Delta F = -4.9$ Kcal/mole

As ΔF is negative, the reaction should proceed spontaneously, giving a cell voltage of 0.21 v.

Theoretical Efficiency at Different Wavelengths — Theoretical maximum efficiency of converting radiant energy at various wavelengths into electrical energy can be calculated from the following equation:

$$Q = \frac{100\ (\Delta F^\circ)\ (298)\ (\Phi A)}{E_\lambda}$$

where:

ΔF° = Standard free energy for the over-all reaction in terms of Kcal/mole of product A

ΦA = Quantum yield of product A

E_λ = Energy of light in Kcal/mole of quanta at wavelength

Values are shown in Table 1. At the ultraviolet end of the spectrum, conversion efficiency is much

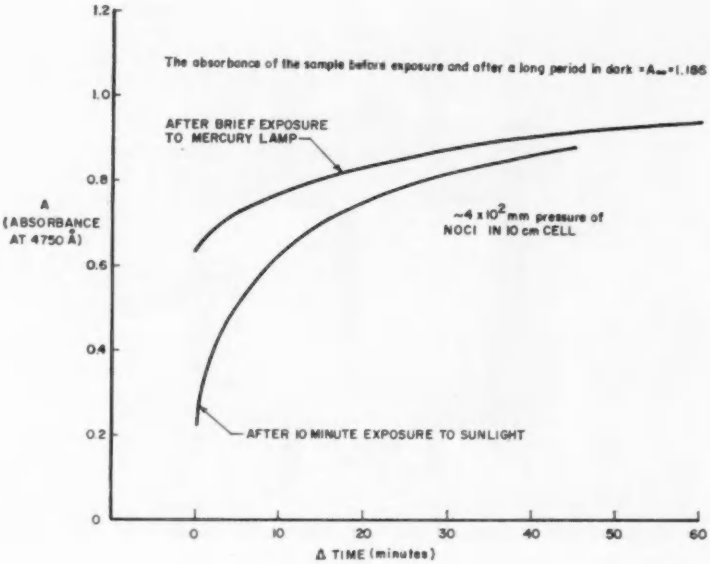

Fig. 1 — **Back reaction from NOCl photolysis.** Cary recording spectrophotometer measured the increase of light absorption caused by the build-up in the nitrosyl chloride content.

Fig. 2 — **Nitrosyl chloride one-cell fuel cell schematic.**

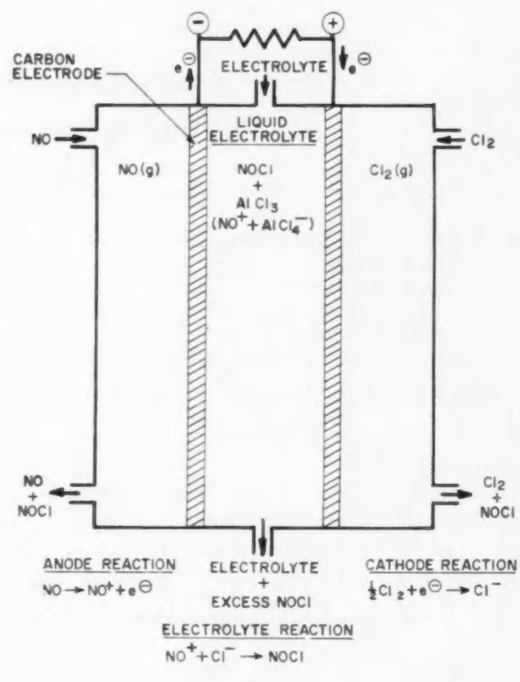

Table 1 — Theoretical Conversion Efficiency

Wavelength, A	Efficiency, Q
6500	22.3%
3000	10.3%
2000	6.9%

lower. . . . Quanta absorbed have far more energy than is required for dissociation of nitrosyl chloride. So, the excess energy is wasted as heat.

Maximum Power Obtainable — The following equation can be used to calculate the maximum power available from a nitrosyl chloride fuel cell which is being regenerated by one square meter of absorber surface:

$$P = (1.6 \times 10^{-13})\,\Phi IaNV$$

where:

P = Power, microwatts/cm²
Φ = Quantum yield
Ia = Light intensity in quanta absorbed/sec × cm²
N = Voltage of the chemical cell
V = Number of equivalents/mole

However, a number of assumptions — which may not be completely correct — must be made.

1. All light up to 6300 A is absorbed only by NOCl,

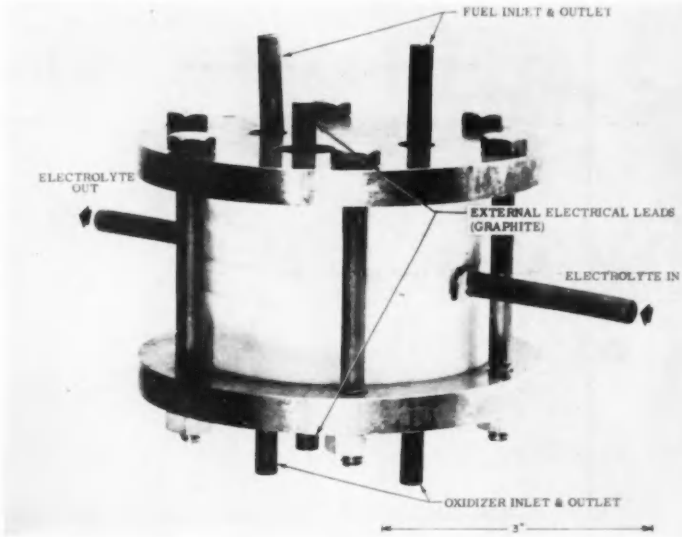

Fig. 3 — **Nitrosyl chloride fuel cell,** which operates at room temperature and pressures up to 100 psig.

Solar regenerative fuel cell system

. . . continued

not the Cl_2 or NO. This amounts to an *Ia* value of 141×10^{15} quanta/cm².

2. $\Phi = 2$ over the range.
3. No back reaction occurs.
4. Theoretical voltage is 0.21 v.

The equation then yields $P = 95$ w/m².

Overall Theoretical Efficiency — Over-all maximum efficiency of converting solar energy into electrical energy is calculated as follows:

$$\% \text{ Efficiency} = \frac{\text{chemical power available from conversion}}{\text{power from incident solar radiation}} \times 100$$

For the range 2150A–6300A, incident power is about 560 w/m². Therefore:

$$\text{Efficiency} = \frac{95}{560} \times 100 = 17\%$$

Considering the entire solar spectrum (1400 w/m²), efficiency is about 7%.

Photodissociation test results

Photodissociation of nitrosyl chloride can be carried out in either gaseous or liquid state or as a solution in an inert solvent.

The gaseous state gave best results because its back reaction is slowest.

Preliminary tests of liquid nitrosyl chloride showed it had several important advantages. It was more compact than the other systems . . . only a ¼-in. absorbing layer was needed, compared to several inches for gas. It also had a built-in separation device . . . nitric oxide produced during irradiation was insoluble and would bubble off as gas leaving the chlorine behind in the liquid.

Using a flow system — with separation attempted within a second of irradiation — nitric oxide was separated as predicted. But, working at the high pressures required, back reaction rate was so rapid that there was only a very small yield.

A few preliminary runs in which nitrosyl chloride dissolved in various halogenated solvents was irradiated showed that the recombination rate was too fast in this instance also.

Gaseous irradiation proceeded satisfactorily. For example, after short irradiation with a mercury lamp, nitric oxide was separated in amounts up to 10% of the nitrosyl chloride present. Up to 80% was decomposed by 10-min exposure to sunlight. . . . However, no attempt was made to separate the products in this instance.

Fig. 1 shows the rate of recombination of photolytically produced nitric oxide to form nitrosyl chloride. It can be seen that there will be a reasonable time for separation to be effected.

Separation of vapor phase products will be more cumbersome than from a liquid irradiation. However, several methods are available — including compression and condensation, absorption by an inert solvent, or absorption of chlorine by a reactive solvent from which it can be later separated. Some recombination will probably occur in all these processes.

Production of power from photolysis products

Direct conversion of chlorine and nitric oxide on the electrodes of a fuel cell removes the Carnot efficiency limitation, and makes possible the use of a low-energy reaction which might otherwise be impractical.

This is the first time that a fuel cell operating on

these fuels has been attempted. Liquid nitrosyl chloride is unique in its ability to act as an ionizing solvent for the fuel cell electrolyte. Its action is somewhat like that of water in the hydrogen-oxygen fuel cell.

Fig. 2 is a simplified schematic of a one-cell fuel cell operating on gaseous chlorine and nitric oxide. The electrolyte is aluminum chloride which reacts with nitrosyl chloride to form conducting ions as follows:

$$NOCl + ALCl_3 \rightarrow NO + AlCl_4^-$$

Fuel cell reactions are: nitric oxide ionizes at one electrode by giving up an electron and becoming NO; chlorine picks up an electron at the other electrode and becomes Cl^-. The chloride ion immediately reacts with the excess NO ions to form the solvent, un-ionized NOCl.

Because of the low boiling point of the electrolyte — less than 0 C — early work on these fuel cells was carried out at - 10 C to - 20 C. Thus, it was possible to maintain a liquid electrolyte at atmospheric pressure.

Fuel cell reaction was established; theoretically-predicted voltage of 0.21 v per cell was obtained. But — currents were extremely low.

In order to operate at room temperature and pressures up to 100 psig, the rather massive cell shown in Fig. 3 was constructed. (Internal construction is similar to Fig. 2.)

High vapor pressure of electrolyte — in addition to its corrosiveness — caused substantial operating difficulties. It had to be pressurized to at least 30 psig to remain liquid at room temperatures. Pressure regulation was devised to sense the pressure over the electrolyte and supply nitric oxide and chlorine at approximately equivalent pressure.

Tests run on this high-pressure, room-temperature fuel cell produced the theoretical 0.21 v per cell, and a very encouraging current output. At short circuit, a current density of 7.5 amp/ft^2 was obtained for a short time before polarization reduced it . . . and at 0.1 v, approximately 1.5 amp/ft^2 was obtained.

Now it is possible to concentrate on increasing current output through various improvements in electrode catalysis and conductivity.

System development

Incorporation of the two separate phases of the study — solar regeneration and fuel cell — into a single unit will allow utilization of nitric oxide and chlorine produced by solar regeneration.

Fig. 4 is a functional schematic of such a system. Light is absorbed into the regenerator where part of the gaseous nitrosyl chloride is decomposed into chlorine and nitric oxide. The gases then pass into the scrubber where chlorine and unreacted nitrosyl chloride are absorbed in a nonvolatile solvent. Nitric oxide passes through to storage or the fuel cell.

In the stripper, heat is applied to drive chlorine and nitrosyl chloride out of the solvent. These gases then pass to storage or to the fuel cell.

In the cell, nitrosyl chloride is produced. Excess nitrosyl chloride — which dissolves in the electrolyte — is removed in the evaporator and sent to the regenerator. The concentrated electrolyte passes back to the fuel cell.

To Order Paper No. 179C . . .
from which material for this article was drawn, see p. 6.

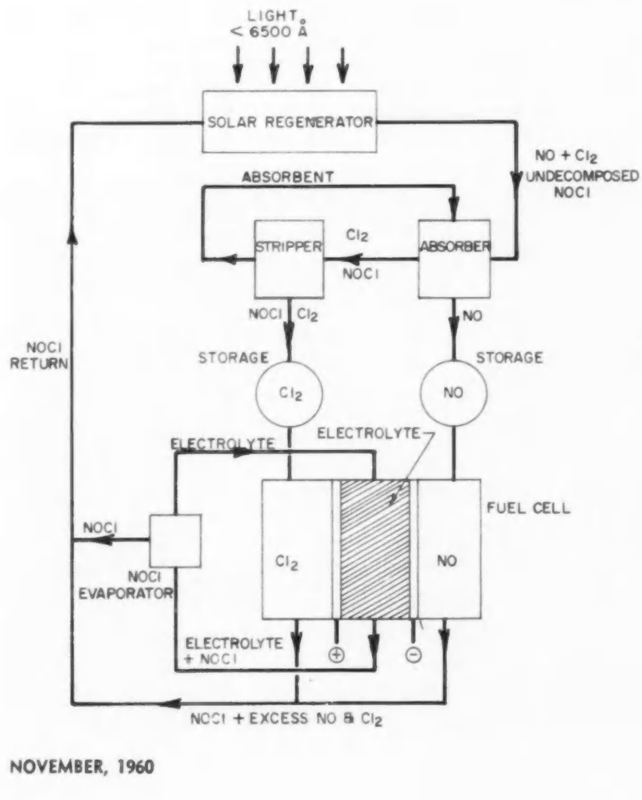

Fig. 4 — **Solar regenerative fuel cell system.**

Axle Gear Wear Cut by SAE

. . . switch from SAE 90 also lifts load-carrying ability

Based on paper by

W. A. Johnson

Rockwell-Standard Corp.

AN SAE 140 VISCOSITY lubricant is far superior to an SAE 90 oil in its ability to prevent axle gear wear, Rockwell-Standard test findings indicate. In many cases wear problems have been overcome merely by changing from an SAE 90 to an SAE 140 lubricant.

These and other findings suggest that rather than seeking a common lubricant for a commercial vehicle transmission and rear axle, economics makes preferable lubricants tailored for each component.

Sufficient field tests have been run to convince R-S engineers that the SAE 140 oils do not run hotter than the SAE 90 oils. In fact, a tendency appears for the SAE 140 lubricant to run 5–10 deg cooler. So, R-S engineers are satisfied that the dynamic viscosity of the SAE 140 oil is higher than that of the SAE 90.

Lab confirms field tests

Laboratory data support the field test data in proving the superiority of SAE 140 in preventing gear wear. Such supporting data resulted from the test procedure shown in skeleton form in Table 1. Each drive unit was run in successive stages. The first three stages represent the break-in, and subsequent stages the test proper. The gears are inspected for scuffing following each stage after break-in. A total of 18 drive units was tested, three each with the two lubricants, and at the three speeds indicated.

Results of these tests are shown graphically in Fig. 1. Actual data are limited to three points, but curves are drawn, nevertheless. At 3000 rpm the

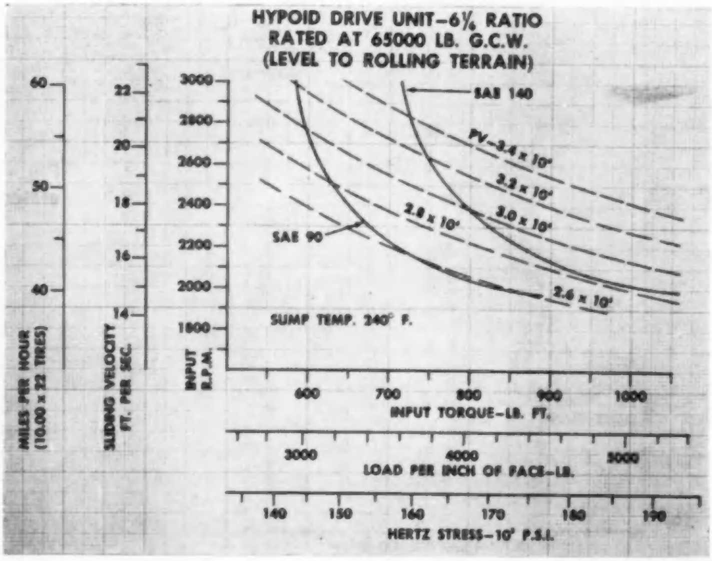

Fig. 1 — In this graphic presentation of gear scuff versus lubrication viscosity tests, conversion scales are provided as well as five constant *PV* curves for reference purposes.

140 Lube

Table 1 — Test Procedure for Gear Scuff versus Lubricant Viscosity

Stage	Input Rpm	Torque	Duration
1	750	262	67 min
2	1500	262	34 min
3	2000 2250	262	22 min
4	2000 2500 3000	262	2 hr
5	2000 2500 3000	306	2 hr
6	2000 2500 3000	350	2 hr
7	2000 2500 3000	394	2 hr
8	2000 2500 3000	438	2 hr
Etc.			

Drive unit water cooled (as required) to maintain sump temperature at 240 F. Gears inspected for scuffing at completion of stage 4 and each subsequent stage until termination of test.

Fig. 2 — Pinion tooth faces show approximately ⅛-in. wear in this typical failed gearset. Failure ceased with use of SAE 140 viscosity, CRC 10-level GL-4 lubricant.

Table 2 — Special Fatigue Test at 45% Full Load — Single-Reduction Axle, Spiral-Bevel Gearset with 63-Deg Angle Drive

(Ratio 5.167 — input torque 1973 lb-ft — input rpm 243-load per inch of face 6164 lb — Hertz stress 239,000 psi-sliding velocity 1.99 fps — Lube SAE 140 viscosity)

Test No.	Gear Material	Lube Type	Lube Tempera-ture, F	Pinion, Cycles	Final Condition — Pinion
4390	Standard	MIL L-2105	250	550,152	1 tooth cracked — rest pitted & scuffed
4392	Standard		250	950,130	1 tooth flaked off — rest heavily flaked
4400	Standard		250	568,620	1 tooth heavily flaked — rest heavily pitted
			Average	689,634	
4377	Standard	10 Level GL-4	250	1,000,188	Bearing rolls flaked — teeth lightly scuffed
4426	Standard		250	750,141	Bearing rolls flaked — teeth lightly scuffed
			Average	875,164	
4377	Continued	6 Level GL-4	250	2,599,808	Teeth flaked & scuffed
4426	Continued		250	950,121	Teeth flaked & scuffed
			Average	1,774,464	
4519	Special	MIL L-2105	250	1,050,060	Teeth pitted & scuffed
4524	Special		250	2,300,100	Teeth scuffed & rippled
4525	Special		250	1,250,090	Teeth pitted & scuffed
			Average	1,533,417	
4588	Special	6 Level GL-4	250	1,400,166	Pinion teeth scuffed & pitted
4595	Special		250	1,050,003	Pinion teeth scuffed & pitted
4597	Special		250	1,950,240	Gear tooth surface cracked — pinion teeth scuffed
			Average	1,466,803	
4484	Standard	MIL L-2105	185	1,800,144	Teeth flaked & scuffed
4485	Standard		185	1,648,998	"Beam" failure on gear teeth
			Average	1,724,571	
4486	Standard	6 Level GL-4	185	1,400,166	Teeth flaked & scuffed
4487	Standard		185	1,300,050	Teeth flaked & scuffed
			Average	1,350,108	

Fig. 3 — Typical gearset of production steel run with MIL-L-2105 lube at 250 F. Pinion teeth have pitted at pitchline and are beginning to flake progressively. One tooth is cracked and failure is imminent.

Fig. 4 — Production steel gearset from test 4377 (Table 2) after running first part of test with 10-level GL-4 lube. Gears are still in good condition.

SAE 140 lubricant supported a torque load 21% greater than the SAE 90 oil. The advantage was 23% at 2500 rpm and 20% at 2000 rpm.

Load-carrying ability

The load-carrying ability of a lubricant is a factor warranting attention aside from the influence of viscosity. In this respect field experience shows some of the older multipurpose lubricants, often referred to as MIL-L-2105 types, perform satisfactorily in all applications while some others are unsatisfactory in many applications. Then there is an in-between group, which may perform well in one application and poorly in another. Some perform erratically in the same application, some units operating without trouble, others with occasional failures, and still others with repeated failures.

Erratic performance is exampled by a fleet of 40 trackless-trolley coaches, which had 147 hypoid gearset failures within a two-year period. Eight coaches had no failures, one had eight, and another had ten. The accumulated mileage for each coach was about 120,000 miles. Fig. 2 shows a typical failed gearset. The pinion tooth faces have worn approximately 1/8 in.

Trackless-trolley service is severe because it involves almost continuous start and stop. Moreover, the drive torque is taken on the normal coast side of the teeth, while the dynamic braking torque is taken on the normal drive side of the teeth. The operator diagnosed the problem as metallurgical; we claimed it was due to improper lubrication. Various types of gears were tried but failure continued. Then it was agreed to try a different lubricant and, with the use of GL-4 lubricant having a CRC 10-level treatment and an SAE 140 viscosity, the gear failures ceased dramatically.

Special fatigue test

Table 2 shows the results of dynamometer tests of a transit coach axle. This was a single-reduction axle having a spiral bevel gearset with the input shaft at a 63-deg angle instead of the usual 90. The test duplicates the maximum driving torque available in the coaches, but applies this maximum constantly, whereas in the coach the torque is applied only once for a short period at the beginning of each start. Thus, the test represents an accelerated condition but duplicates actual field failures.

MIL-L-2105 lubricant was taken as the yardstick because it has a good field performance record. The tests were to have been run with GL-4 lubricant, with one having a 10-level treatment, but through a mixup in communications most of the tests were run with a 6-level GL-4. The second group of tests (4377 and 4426) were run in two parts because imminent failure of the pinion bearing required a stop for replacement. Valid comparison of the first two groups of tests using production steel and a sump temperature of 250 F are lacking because of the original error in use of lubricants. The last four groups of tests indicate the 6-level GL-4 lubricant to be less effective than the good MIL-L-2105 type,

. . . continued

Table 3 — Results of Special Fatigue Test

(Hypoid drive unit — 6 1/6 ratio — rated at 55,000-lb GCW-input torque 2110 lb-ft — load per inch of face 12,070 lb — Hertz stress 304,000 psi — input rpm 463 to 617,000 pinion cycles and then 617 to failure — sliding velocity 3.16 fps at 463 rpm and 4.21 at 617 rpm — lubricant SAE 140 viscosity — sump temperature 250 F)

Test No.	Lube Type	Pinion, Cycles	Final Condition — Pinion
4596	MIL L-2105	817,241	1 tooth flaked & cracked — all teeth scored
4606		527,820	Teeth scuffed & pitted
4607		752,456	Teeth scuffed & pitted
	Average	699,172	
4608	6 Level GL-4	400,495	Pinion bearing failed — teeth starting to scuff
4609		632,141	Teeth pitted and starting to scuff
4610		766,030	1 tooth cracked — all starting to scuff
	Average	699,085[a]	

[a] Does not include test 4608.

especially when the sump temperature is held at 185 F. Figs. 3 and 4 are illustrations from the fatigue tests.

The results of an over-the-road test of a hypoid drive unit are shown in Table 3. The test was run in three stages. Stages 1 was the break-in; stage 2 a run to 617,000 cycles; stage 3 to failure. The gears were inspected for distress every 50,000 pinion cycles during stages 2 and 3. In this case the lubricants appear equally effective. Fig. 5 and 6 illustrate some results from this test.

Fig. 5 — Gearset from test 4596 (Table 3) using MIL-L-2105 lube. Teeth are heavily scored and pinion teeth are flaking. One tooth is cracked and complete failure looms.

Tailored lubricants best

Most recommendations for transmissions specify an SAE 90 viscosity without load-carrying additives or at most a mild load-carrying additive treatment. This recommendation, in our opinion, is dictated by factors other than lubrication. For example: R-S markets axles having single-reduction hypoid-drive units and planetary secondary reductions in the wheel ends. For these axles use of an SAE 140 lubricant in the housing bowl for the hypoid drive unit is recommended, but an SAE 90 is specified for the wheel-end reductions because it is more readily put into and taken out of the planetary pinion bore. Undoubtedly, wheel-end design could be changed to accommodate the SAE 140 lubricant or a larger drive unit could be used to accommodate the SAE 90 in the housing bowl, but either change would raise the cost of the unit.

If necessary, the same lubricant could be used for transmission and rear axle by derating the axle and using an axle one or two sizes larger. Indeed, it might be possible to use the engine lubricant in both transmission and rear axle if both components were oversized to compensate.

Since actual practice is in the opposite direction, it is believed that overall economics favor having different lubricants, each tailored to the requirements of the engine, the transmission, and the rear axle.

Fig. 6 — Gearset from test 4608 (Table 3) which ran with 6-level GL-4 lube and was stopped because of pinion bearing failure. Gear teeth are still in fairly good condition.

To Order Paper No. 192A . . .
from which material for this article was drawn, see p. 6.

Designing an

. . . for internal combustion engines

of the valve train resonant frequency in

and gas dynamics and force-stress limits in

Based on paper by

J. H. Nourse, R. C. Dennis, and W. M. Wood

Willys Motors, Inc.

OPTIMUM CAM DESIGN for an internal combustion engine is that profile which provides an accurate control of the valve dynamics over a specified range of engine operating speeds, and which also maximizes the area under the valve lift diagram for a specified valve event.

The major cam design parameters are five in number — valve event length, valve lift, engine speed range, valve train resonant frequency, and force-stress limits — of which only three may be chosen independently.

In overhead valve (pushrod) engines the optimum profile may be derived if the engine speed is fixed as a function of the valve train resonant frequency, the valve event length and force-stress limits are arbitrarily specified, and the valve lift is maximized as part of the cam design solution.

In overhead cam (non-pushrod) engines, the valve train resonant frequency does not directly define the engine speeds and, therefore, the engine speed may be chosen based on gas dynamics considerations. In this manner, the five interdependent variables are reduced to three as before, with the problem solution obtained as in pushrod engine design procedures.

General objectives

A qualitative statement of the cam design problem can be given as: based upon a given, desired engine performance (that is, specific torque and horsepower characteristics diagrams), and a particular valve train configuration, how may the optimum cam lobe contour be selected from the infinite number of contours available? The optimum cam lobe contour is that contour which most nearly obtains the engine performance objectives while taking account of the limitations imposed upon the design by the elastic characteristics of the engine system.

The single, most important rule for obtaining a successful cam profile is that of starting the design with objectives which are both reasonable and compatible with the design limiting criteria. Engine design performance objectives will seek to specify (cam design) values for:

1. Valve event (and timing) — controlled over a specified range of engine speeds up to some maximum speed.
2. Valve lift — controlled over a specified range of engine speeds up to some maximum speed.

The limitation to the accomplishment of any such arbitrary set of goals will be fixed by certain physical characteristics in the valve train. These may be broadly divided into two classifications:

1. Valve train dynamics limits — where, in overhead-valve (push-rod) systems, the fundamental natural resonant frequency of the valve train will normally limit the practical maximum engine operating speeds, and
2. Force-stress limits — wherein the valve train interface designs, including metallurgy and lubrication, will fix the maximum allowable forces and bearing stresses. These limits will normally establish the maximum allowable accelerations in cam designs for either overhead-cam or overhead-valve engines.

The important point is that we seek a particular valve motion to obtain a desired engine performance; from the cam profile design point of view, the valve train system characteristics limit the performance that can be realized. The relationship between the parameters affecting design goals as opposed to systems limitations are both numerous and subtle. If the cam profile is to be optimized, these parameters must be defined quantitatively. It is necessary to examine the solution boundary conditions to determine which must be considered rigid, and which may be adjusted in favor of some other more desirable requirement. In this engineering problem, as in most, the final solution is the result of compromise. We seek the optimum compromise.

Optimum Cam

requires consideration

overhead valve engines,

overhead-cam engines.

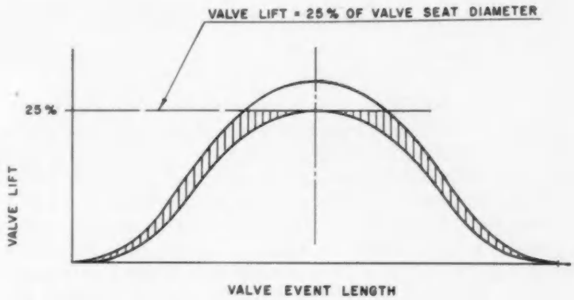

Fig. 1 — Gain in effective valve lift diagram area with optimized lift.

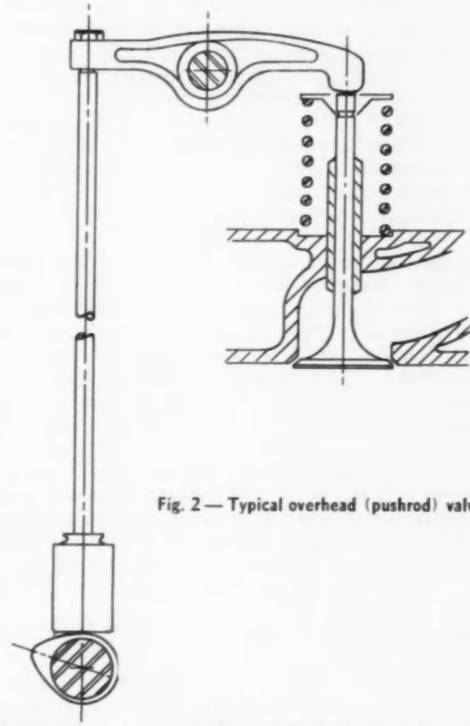

Fig. 2 — Typical overhead (pushrod) valve train.

Specific design goals

Consider first, the specific values of valve event length, valve lift, and operating range of engine speeds as these relate to the gas dynamics problem of engine design performance.

Valve Event Length — In the context of this problem, the subject of gas dynamics relates the particular configuration of the engine gas flow passages (with valve timing and event lengths) to the desired characteristics of engine performance. While this is a vitally important problem, there exists no analytical method for choosing optimum valve event and timing values expressed as a function of engine manifold geometry for engine torque and horsepower performance characteristics. Fortunately, there does exist an extensive practical experience which allows near optimum values to be chosen empirically. These, normally are then varied in a few carefully chosen trials to maximize a desired performance result. Such trials require a limited number of cam designs, but it is important to recognize that the valve event lengths and timing must be rigidly determined in each trial as well as in the final choice of such values. For the optimum cam design the valve event length, having been selected becomes invariant.

Valve Lift — While it is customary (with engine cam design methodology) to consider the valve lift as a fixed design specification, there are important advantages which will accrue to the cam profile (and to the engine performance) if the design value of the valve lift is not rigidly prescribed. The optimum cam design specification should state instead a minimum and maximum acceptable valve lift limit. The design goal will then be to achieve the maximum lift obtainable within the specified tolerance.

The valve lift required to produce an annular valve-open area just larger than the port area at the valve seat is a lift equal to 25% of the valve seat diameter. The design value for the valve lift is, therefore, normally fixed at this level. There are two reasons why this value should not be a rigid specification.

First, consider the parameter used to evaluate and maximize the engine breathing, the area under the valve-lift diagram. Whit it is true that area added to the diagram above the normal (25% of valve seat diameter) valve lift level is relatively ineffective, there is a significant gain in area below this level associated with a higher valve lift (Fig. 1). The gas dynamics of the flow through both intake and exhaust valves show that a true evaluation factor for the relative effectiveness of the areas under the valve-lift diagram would be quite complex but that, however this may be done, a practical and simple maximizing of the area achieves the desired effectiveness evaluation result. The important point is that higher valve lifts improve the effective breathing of the engine and also that this effectiveness can be evaluated from the valve lift diagram area even though the gain in effectiveness with area is not linear.

Second, the design values for valve event, valve

Designing an Optimum Cam

. . . continued

lift, engine design speed, valve train resonant frequency, and the maximum allowable interface bearing stress are interdependent quantities in any particular cam design. The maximum possible valve lift attainable (assuming there is no clearance problem) will, in general, increase with a longer valve event, increase with a higher valve train resonant frequency, and also increase with a higher allowable stress, but will decrease with a higher engine design speed. The interdependency of these factors is such that it can, and occasionally does, happen that the specified value for valve lift cannot be obtained without exceeding one or more of the limiting factors. An important part of the cam design problem is the discovery of this fact; but equally important is the discovery that a greater valve lift is available to a given design. For example, it is usual to gain 5–10% in valve lift area by optimizing the valve lift; in a recent cam solution, an improvement in engine peak horsepower performance of 16% was obtained from this single consideration.

Engine Speed Range — The third (and last) cam design factor involved with gas dynamics is the range of engine design operating speeds. While torque performance depends upon an accurate valve timing and control related to single units (cylinder volumes) of fuel-air mixture in each cylinder, the horsepower performance is related to the rate at which successive units of mixture can be effectively processed. That is, power performance is closely identified with engine speed. Since accurate valve control over a range of engine speeds depends upon the valve train resonant frequency more than any other single factor, from the point of view of cam design, the power performance, the range of operating speeds in general, and the maximum engine operating speed in particular, depend primarily upon the stiffness and mass (that is, the resonant frequency) of the valve train.

We have broadly classified into two parts the limiting factors in a cam design; that is, the valve train dynamics and force-stress limits. The importance of the valve train resonant frequency parameter can best be appreciated by comparing the difference in the range of engine speed performance between the overhead-valve and the overhead-cam type engines. The pushrod engine will normally have a valve train resonant frequency of 30,000–50,000 cpm; the corresponding maximum engine speeds will be 4000–6000 rpm. These ranges assume that the valve spring load at the cam will be low enough that excessive bearing stresses will not occur at engine crank and idle speeds with normal production materials and lubrication. In the overhead-valve engine, the valve train resonant frequency will design the maximum engine speed; the force-stress limits will design the maximum allowable cam profile accelerations.

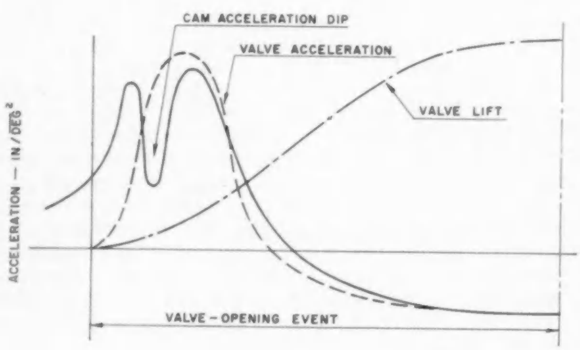

Fig. 3 — Acceleration diagram showing cam dip.

The overhead-cam engine will normally have a valve train resonant frequency of 200,000–500,000 cpm and a maximum engine speed of 6000–8000 rpm. On the basis of valve train frequency, the engine speed would not become important below about 12,000 rpm; the force-stress limits are the controlling factor in the overhead-cam engine cam design.

Consider the general arrangement of a typical overhead-valve gear design (Fig. 2) and keep in mind that the specific function of the gear is to control precisely the motion of the end element (valve) at engine speed. As the tappet moves up the cam opening ramp, first any clearance, then the static and dynamic deflections in the valve train, are taken up in such a way that, ideally the valve will just lift from the valve seat as the tappet reaches the top of the ramp. At this moment, then, the valve train has experienced some compression and has stored an amount of elastic energy.

In the first few degrees of the cam opening event, much larger acceleration forces are applied by the cam to the valve train to accelerate the valve. Further valve train compression takes place, and thus still more potential elastic energy is stored in the valve train system. If at this moment the accelerations imparted by the cam continue, the system will rebound with the potential energy of compression exchanged for kinetic energy in the end (valve) element. The effect at the valve is to induce suddenly a much higher acceleration than that given by the cam. The elapsed time required for this rebound to occur is precisely related to the natural resonant frequency of the valve train system, and it is important to note that this interval is wholly independent of engine speed.

In the polydyne system of cam design, the acceleration forces applied to the valve train system at the cam are adjusted in such a way (that is, in the right amount and at the right time) so as to just relieve and compensate for this release of potential compression energy. By this means, the valve acceleration diagram is properly controlled; while the cam acceleration diagram shows a pronounced dip (Fig. 3). Because, however, the elapsed time for valve train compression and rebound depends directly upon valve train frequency and is independent of engine speed, the compensation in the cam profile can be exactly positioned for only one engine speed — this will be the engine design speed.

Thus, the optimum cam design goal for engine speed range will be fixed by a consideration of the valve train resonant frequency in overhead-valve engines — and by a consideration for gas dynamics and force-stress limits in overhead-cam engines.

To Order Paper No. 202A . . .
from which material for this article was drawn, see p. 6.

Bottling up Outboard Engine Noise

Based on paper by

James W. Mohr

Outboard Marine Corp.

KILLING the noise of an outboard engine takes four engineering steps:

- Bottling up the noise of the engine.
- Silencing the combustion air inlet.
- Suppressing the exhaust noise.
- Sealing the holes used for control, propulsion, and servicing.

The first of these steps is a double one in which two types of noise must be contained; that directly audible, and the vibration of the engine which is picked up by the boat and which could be defined as a "shaking" noise. The first part is taken care of by putting a housing or shield around the engine. The second part requires a dynamic vibration isolation system.

A schematic system for meeting these objectives is shown in Fig. 1. Here, the cover is considered a part of

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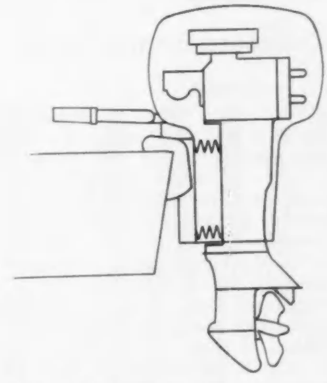


Fig. 1 — Schematic diagram of motor isolation from boat and cover.

Engine Configuration	Primary		Secondary	
	Shaking Force	Rocking Couple	Shaking Force	Rocking Couple
Single Cylinder	$\frac{WrN^2\cos\theta}{35250}$ Fore Aft Plane	0	$\frac{Wr^2N^2}{35250\ell}\cos 2\theta$ Fore Aft Plane	0
Twin Cylinder	0	$\frac{aWrN^2}{35250}\cos\theta$ Fore Aft Plane	$\frac{2Wr^2N^2}{35250\ell}\cos 2\theta$ Fore Aft Plane	0
V4 Cylinder	0	0	$\frac{2\sqrt{2}\,Wr^2N^2\cos 2\theta}{35250\ell}$ Lateral Plane	0

W = Weight of Reciprocating Parts for 1 Cylinder - Pounds
r = Crank Radius - Inches
N = Engine Speed - RPM
θ = Crank angle from TDC - Degrees
ℓ = Connecting Rod Length - Inches
a = Distance Between Cylinders - Inches

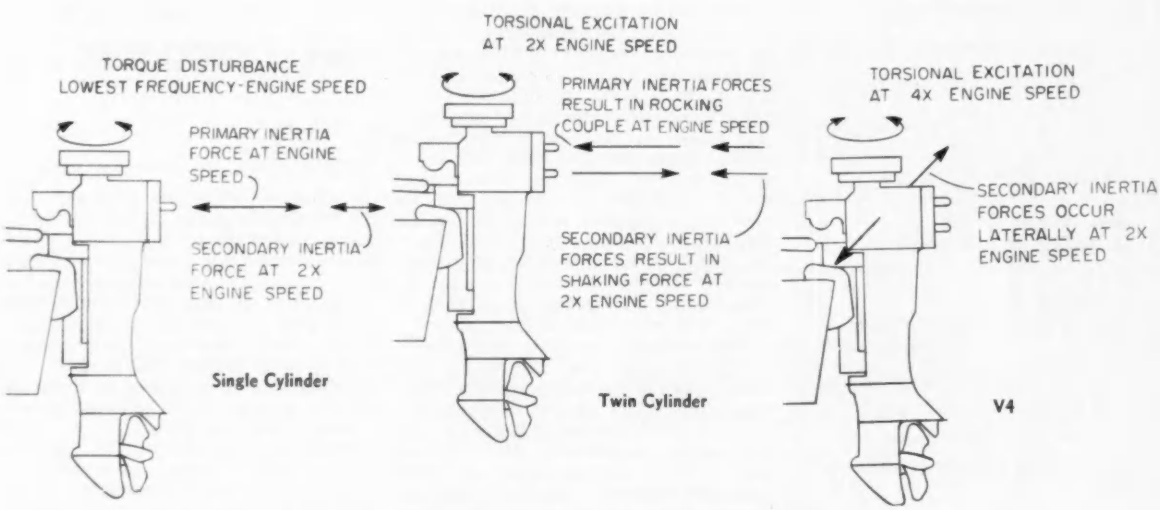


Fig. 2 — Inertia forces and rocking couples for balanced outboard engines.

Bottling up Outboard Engine Noise

. . . continued

the boat and the engine is suspended dynamically within the cover. This system not only suppressed engine-radiated noise but also makes it possible to use a simple system to isolate vibration.

The three types of force systems that have to be isolated are for single, twin, and V-4 cylinder arrangements. A chart of the driving forces and schematics of their application are shown in Fig 2. Isolation of these forces from the boat requires that the natural frequency of the engine on its suspension system not coincide with the frequency of an engine's driving forces.

One approach to this problem is the mounting system (simplified) used by OMC for its twin engine. (See Fig 3.)

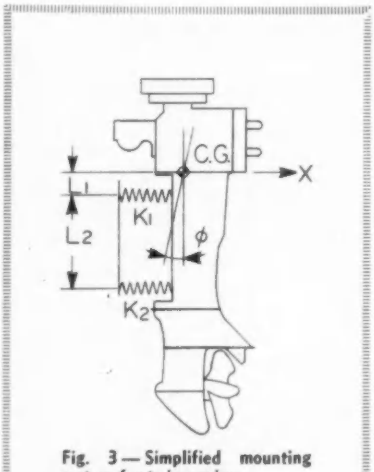


Fig. 3 — Simplified mounting system for twin engine.

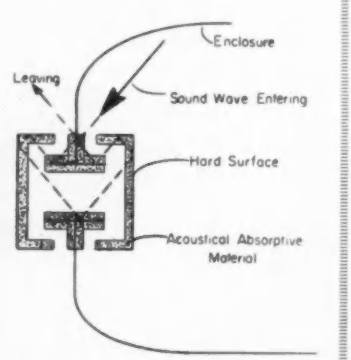


Fig. 4 — Sound trap for combustion air inlet.

There are two natural frequencies, which are given by:

$$f_{n_1}{}^2 = \frac{60}{2\pi} \times \frac{1}{2}\left(\frac{C}{r^2}+a\right) - \frac{1}{4}\sqrt{\left(\frac{C}{r^2}-a\right)^2+\frac{b^2}{r^2}}$$

$$f_{n_2}{}^2 = \frac{60}{2\pi} \times \frac{1}{2}\left(\frac{C}{r^2}+a\right) + \frac{1}{4}\sqrt{\left(\frac{C}{r^2}-a\right)^2+\frac{b^2}{r^2}}$$

$$a = \left(\frac{K_1+K_2}{W}\right) g$$

$$b = \left(\frac{K_2L_2-K_1L_1}{W}\right) g$$

$$c = \left(\frac{K_1L_1^2+K_2L_2^2}{W}\right) g$$

where:

f_{n_1} = Natural frequency at 1, cpm
f_{n_2} = Natural frequency at 2, cpm
W = Weight, lb
g = 386 in./sec/sec
L = Distance, in.
K = Spring rate, lb/in.
r = Radius of gyration about cg, in.
K_1 = Total spring rate at 1
K_2 = Total spring rate at 2
L_1 and L_2 are plus if cg is between K_1 and K_2
L_1 and L_2 are plus if cg is between K_1 and K_2
L_1 is minus if cg is above K_1
L_2 is minus if cg is below K_2

The system is decoupled if either:

$$L_1 = L_2 \text{ and } K_1 = K_2 \qquad (1)$$

or:

$$K_1L_1 = K_2L_2 \qquad (2)$$

This means that the rotational and translational modes of vibration are independent. The advantage of decoupling is that translational resonances do not occur at rotational natural frequencies, and vice versa.

In outboard engines, there are only two primary conditions that have to be avoided, full power and idle speeds. Of these two, the full power is the most important since it accounts for about 90% of the engine usage and the driving forces at full power are much greater than at idle, since the force varies as the square of the engine speed. Therefore, the designer has to pick his suspension springs constants and distances, so at best there only will be transient operation at resonance speeds. This can usually be done so that the effectiveness of the isolation is 90% (transmissibility is 0.1).

Torque variations must also be isolated from the boat. These are the most severe at low engine speeds. The same approach is used in that the torsional natural frequency is found and a suspension isolates it. An effectiveness of 90% can be easily achieved since the natural rotational frequency is well under the idle speed (for twin engines).

Cutting inlet air noise

An acoustical baffle system can be used to get air into the engine compartment but not let disturbing sound out of it. Since the whole engine enclosure acts as a muffler chamber, there is very little problem in making it silence the incoming air. All that remains is to provide a sound trap at the entrance to prevent high-frequency engine-radiated sounds from escaping. Such a trap is shown in Fig. 4.

The sound trap has a low pressure drop for entering air but makes existing sound waves bounce off many surfaces before they escape. If these surfaces are covered with a noise absorbing material, the sound will be drastically reduced.

Exhaust noise

This is the simplest problem since the exhaust is ported under water. One point must be watched though, the exhaust must be entrained in the slipstream of the propeller. Otherwise, the exhaust gas forms a channel to the surface through which the noise can pass.

Sealing the enclosure

Rubber seals are used to surround the flexible control lines and line the edges of covers used for maintenance. A soft rubber seal is used on the lower end of the enclosure. Since it isn't always desirable to seal at the point of least relative motion between the engine and housing, these seals may have to withstand deflections of around ¼ in.

To Order Paper No. 183B . . . from which material for this article was drawn, see p. 6.

New Way to Adhesive Bond at 3000-6000 F

Based on paper by
ROGER A. LONG
Narmco Industries, Inc., Telecomputing Corp.

A NEW method of inorganic adhesive bonding is based on the ability of various carbides or borides to form solid solutions and eutectics when held in contact at elevated temperatures for some time. It may be the only type of process applicable for very high-temperature structural applications ranging from 3000 to 6000 F.

It can best be explained by examining its application to a specific research project, as follows:

The project problem is to join graphite to tungsten metal in which the bond line must withstand 4500 to 5000 F. In this project, the tungsten

continued on p. 132

SAE NEWS

JAMES MARK CRAWFORD, president of the Society of Automotive Engineers in 1945, died on Friday, Sept. 23, 1960 in La Jolla, Calif., where he had made his home since retirement from General Motors Corp. in 1951.

His engineering career was marked by coverage of a variety of assignments which called for successful application of an unusually wide range of aptitudes. He first gained industry recognition as chief engineer of a small car company (Auburn), where personal engineering talent applied directly to design of the product was of major importance. Executive problems of organization were minor. Then, at the end of his active engineering career, he was carrying top executive responsibility for the General Motors' engineering organization, one of the largest in the automotive industry.

Forthrightness was one of his outstanding characteristics. A keen analyst, he was always clear about his opinion of any situation once he had obtained the facts. And he was always willing for others to know his thoughts.

He was chief engineer of Chevrolet in 1945, when he was SAE president. Then he went on to become, before his retirement, vice-president — engineering, General Motors Corp.

During his term as SAE president, Crawford saw the SAE Technical Board established. In that same year, he played an important personal part in clarifying SAE's singleness of purpose in its concern with technical and engineering questions. And his active, personal participation was of major importance in reorganization of the working structure of the Coordinating Research Council, of which — with the American Petroleum Institute — SAE is a sustaining member.

Following his retirement, he established his home at La Jolla, Calif., and devoted much time to oil painting. (He had studied magazine illustration at the Chicago Art Institute as a young man, before entering industry.) Shortly, he became a member of the board of directors of the La Jolla Art Center; then was elected president. There he was responsible for considerable improvement in the organization's operating methods, and, in January, 1960 saw completion of a new $700,000 wing of the Center, in the development and financing of which he had played an important part.

He established SAE's James M. Crawford Fund in 1957 "to see the cooperative method of getting things done preserved . . . and matured to even higher levels of development in SAE." SAE Council accepted the fund with the declared intention of devoting it to "enrichment of satisfactions to individual members resulting from their participation in SAE Technical Committee work." To establish the fund, Crawford turned over to SAE securities with an income of approximately $685 per year.

Crawford began his automobile career as a draftsman with the Old American Motor Car Co. in his home town, Indianapolis. Then, after becoming assistant chief engineer, went to Chalmers Motor Co. and Auburn Motor Car Co. In the latter position, he was responsible for Auburn designs during the period of its sensational rise following assumption of the presidency by E. L. Cord.

He joined the General Motors organization as assistant chief engineer of Chevrolet in 1927, when O. E. Hunt was chief engineer.

James Mark Crawford

1886 – 1960

SAE Policy on CRC Is Okayed by SAE Directors

SAE POLICIES governing the Society's participation in the Coordinating Research Council have been established in codified form by action of the SAE Board of Directors.

Based on a report by a special committee of the Board, the policy statement covers such things as:

- SAE financial support to CRC.
- SAE attitudes toward CRC administrative policies.
- Procedures in SAE for maintaining cognizance of CRC operations.
- Responsibilities of SAE nominated directors of CRC.

The committee upon whose recommendations the SAE Directors acted was chairmanned by George J. Huebner, Jr. who is currently president of CRC. Their report recalled that each of CRC's two sustaining members (SAE and American Petroleum Institute) nominates seven directors on the 14-man CRC Board of Directors, each for a two-year term. Terms are so staggered that SAE nominates three directors one year and four the next. The CRC President and Vice-president are elected by the Board of Directors for a two-year term, the offices alternating from among the SAE- and API-nominated Directors. Serving with Huebner on the special committee were W. Paul Eddy, V. G. Raviolo, Arthur Nutt, and Trevor Davidson.

Huebner

As regards SAE financial support of CRC, the newly approved SAE policies say:

SAE financial support of CRC relates directly to the Society's support of its own technical committee operations. SAE policy, as approved on April 11, 1956, is as follows:

I. SAE's financial contribution to the support of CRC is a part of SAE's technical committee operations.

II. SAE's technical committee operations should be supported entirely by income from industry.

III. If, in any given year, the budget for SAE technical committee operations is reduced because of insufficient income from industry during the previous fiscal year, CRC's appropriation should be reduced in the same proportion.

To keep SAE Directors apprised of CRC policy matters, the new statement requires that the ranking officer of CRC who is an SAE-nominated CRC Director shall report annually to the SAE Board of Directors "on the management function of CRC." Further, this same ranking CRC officer will keep the SAE Technical Board informed on CRC's technical activities.

The long-standing practice of SAE's taking responsibility for distribution by sale of CRC reports is reaffirmed, as is the desirability of maintenance of active liaison between the SAE staff and the CRC staff.

CRC Directors nominated by SAE, the newly approved policy statement continues, "will act as individuals in discharging their responsibilities as CRC Directors under New York State law." The statement concludes:

"The SAE recognizes the importance of continuity of membership on the CRC Board. It further recognizes the fact that a complete understanding of CRC operations at the Board of Directors level takes more than two years. The reappointment of CRC Directors to succeed themselves is permissible."

Section Officers Make Strides at 4th Regional Conference

FORTY representatives of six West Coast Sections were welcomed at the Fourth Regional SAE Section Officers Conference by Conference Chairman I. M. Harlow. Held at San Francisco on August 17, the Conference had as guest speakers SAE President Harry E. Chesebrough, Past-President Leonard Raymond, Sections Board Chairman W. F. Ford, and SAE General Manager Joseph Gilbert.

SAE President Chesebrough (seated) listens to I. M. Harlow address the Fourth Regional Section Officers Conference, held at San Francisco on Aug. 17, during the SAE National West Coast Meeting.

Exchange of ideas on how Sections can best serve SAE members at the local level — and how the Society can help Sections do their job — is the objective of these Conferences, Ford told the group. Success of the initial Section Officers Conference held last year in Vancouver set the pace for those which followed . . . and led to this repeat session on the Pacific Coast, he said.

"Three years of progress in one," was how President Chesebrough summed up advances made in SAE's Progress Program . . . pointing to the coming International Automotive Engineering Congress and Engineering Display in January as one such evidence. General Manager Gilbert told of similar progress in other areas of Society operation, such as the Publication Committee's programs to improve current services and to provide new publications to meet member needs. Past-President Raymond, who was a member of the planning group, expressed his pleasure at the Society's progress under the new program.

SAE MEMBERSHIP GROWTH — in quality and quantity — was a main topic of discussion at the Conference. Some foreknowledge of a nonmember's qualifications would be a wise prerequisite to invitation to apply, it was agreed. As another step to implement Grading Committee action, local Section members were given several sug-

gestions on how to help the applicant complete details of his application. Explanation of Grading Committee procedure, and of resulting delays when it is necessary to go back to the applicant for more detailed information were outlined. (Note: As an outcome of this Conference, the SAE Board of Directors and its advisory Membership Grading Committee have agreed to advise Sections as to the basis for actions taken on certain applications from their territories.)

"NORTHWEST ROUNDTABLE" is what Sections in Northwestern United States and Southwestern Canada call the outcome of their Northwest Planning Council. The Council was established to plan a series of meetings where selected authors will be invited to speak on the same topic at consecutive Section meetings in the area, Otto Kirchner told the group. Similar programs in recent years have proven most successful.

SECTION ALLOTMENTS came in for a share of discussion. Ford told of the Sections Board Finance Committee's study of the situation and of its recommendation which led to Board of Directors' action authorizing the Section Board — in its discretion — to appropriate up to $150 to a Section on request by its Governing Board, to improve member or student services locally.

While interest was still high, the Conference adjourned to permit those present to attend technical sessions of the National West Coast Meeting.

FACTS . . .

from SAE literature

ELIGIBILITY for SAE membership is one of the subjects covered in the Membership Committee's new Handbook and Guideposts. Designed for the use of those participating in SAE's Membership-Increase Program, the new booklet answers the questions most often asked of them.

A new "FACTS ABOUT SAE JOURNAL" is off the press. Among other "facts," the booklet shows that there are SAE Journal readers in 100% of all companies making passenger cars, trucks, aircraft powerplants, helicopters, missiles and drones, tractors, diesel engines, and earthmoving equipment.

(Except where a charge is specifically indicated, SAE Journal will be glad to supply on request one copy of any of the pieces of SAE literature described. Address "Literature," SAE Journal, 485 Lexington Ave., New York 17, N. Y.)

SAE NATIONAL MEETINGS

1961

● JANUARY 9–13 • 1961 • COBO HALL • DETROIT

INTERNATIONAL S·A·E AUTOMOTIVE ENGINEERING CONGRESS · EXPOSITION

● March 13–17
National Automobile Week (National Automobile and Production Meetings), The Sheraton-Cadillac, Detroit, Mich.

● April 4–7
National Aeronautic Meeting (including production forum and engineering display), Hotel Commodore, New York, N. Y.

● June 4–9
Summer Meeting, Chase-Park Plaza, St. Louis, Mo.

● August 14–17
National West Coast Meeting, Sheraton Hotel, Portland, Ore.

● September 11–15
National Farm, Construction, and Industrial Machinery Meeting (including production forum and engineering display)
National Transportation Meeting
National Powerplant Meeting
. . . Milwaukee Auditorium, Milwaukee, Wis.

● October 9–13
National Aeronautic Meeting (including manufacturing forum and engineering display), The Ambassador, Los Angeles, Calif.

● November 9–10
National Fuels and Lubricants Meeting, Shamrock Hotel, Houston, Texas

YOU'LL be interested to know . . .

AVAILABLE FOR SECTION AND GROUP MEETINGS is a 5-min, 16-mm, color film-strip *preview* of ICEAE events at Detroit's Cobo Hall in January.

SAE President Harry Chesebrough and ICEAE Meetings Operations Committee Manager Paul Ackerman are the narrators.

Sections and Groups desiring to show the film at their meetings should notify Robert W. Crory at SAE headquarters, who will arrange for delivery. Several Sections are already on the list for a showing.

SOUGHT FOR EXHIBIT in the Science Pavilion during SAE's International Congress and Exposition of Automotive Engineering in Detroit's Cobo Hall in January are:

- New types of vehicles
- Historic vehicles
- Space vehicles
- New materials
- Human and natural resources developments
- Unconventional energy-conversion systems and fuels
- Transportation mobility

ALEX L. HAYNES, chairman of the Committee arranging the Science Pavilion, says that the sought-for exhibits are to supplement the many that have already been offered. Information on available exhibits in the above categories can be sent to Chairman Haynes at SAE headquarters.

SAE SECTION MEETINGS

BALTIMORE

December 8 . . . Melvin M. Gough, senior test pilot. "Looking Ahead in Aviation." Engineers Club, 6 W. Fayette St. Dinner 6:30 p.m. Meeting 8:00 p.m. Special Feature: Annual Ladies Night.

METROPOLITAN

December 8 . . . Col. Donald H. Heaton, NASA, Washington, D. C. "Departmental Problems in Space Exploration." Brass Rail Restaurant, Fifth Ave., between 43rd and 44th Sts., Manhattan. Cocktails 5:30 p.m. Dinner 6:30 p.m. Meeting 7:45 p.m.

MID-MICHIGAN

November 28 . . . Charles Drury, director of reliability, Central Foundry Division, General Motors Corp. "Reliability from the Foundry Viewpoint." Bancroft Hotel, Saginaw, Michigan. Dinner 7:00 p.m. Meeting 8:00 p.m. Special Features: Coffee Speaker and Film.

NEW ENGLAND

December 6 . . . Thomas J. McLernon, general manager, Metropolitan Transit Authority. "Transportation." M.I.T. Faculty Club, 500 Memorial Drive, Cambridge. Dinner 6:45 p.m. Meeting 8:00 p.m. Special Feature: Social Hour 6:00 p.m.

NORTHERN CALIFORNIA

December 1 . . . J. S. Wintringham, research advisor, Ethyl Corp. "Future Automobile Power Plants." Claremont Hotel, Berkeley. Dinner 6:30 p.m. Meeting 8:00 p.m.

ROCKFORD-BELOIT

December 5 . . . K. Schuster, product development engineer, GMC Truck & Coach Division. "Vehicle Air Conditioning." Wagon Wheel Lodge, Rockton, Illinois. Dinner 6:30 p.m. Meeting 8:00 p.m.

SOUTHERN CALIFORNIA

November 28 . . . John S. Wintringham, Research Laboratories, Ethyl Corp. "Potential Passenger Car Powerplants." Rodger Young Auditorium, 936 W. Washington Blvd., Los Angeles. Dinner 6:30 p.m. Meeting 8:00 p.m.

TEXAS GULF COAST

December 12 . . . Harvey M. Cook, manager plant operations branch, rocket test facilities, ARO, Inc. "Missiles and Space." Houston Engrg. & Scientific Society Bldg., 2615 Fannin St., Houston. Dinner 6:30 p.m. Meeting 7:30 p.m.

WILLIAMSPORT

December 5 . . . F. A. Robbins, chief engineer, Koppers Co. "Piston Ring Design & Development (Emphasis on Metallurgy)." Moose Auditorium. Dinner 6:45 p.m. Meeting 8:00 p.m.

Scuttlebutt from Technical Committees

700,000 AMSs SOLD ANNUALLY — Each working day an average of 2,822 Aeronautical Material Specifications flow from SAE Headquarters into aero-spacecraft industry shops. With over 12 million AMSs sold since 1940, their influence on metallurgical practices in the U.S. has been unprecedented. This flow of information is made possible by the 80 engineers who comprise the Aero-Space Council's AMS Division.

At a September meeting, the AMS Division appointed an ad hoc study group to review its present Commodity Committee structure. This group will be led by AMS Division Vice Chairman W. C. Schulte who is also chairman of the Finishes, Processes and Fluids Commodity Committee. Working with him will be G. F. Kappelt, chairman of the Nonferrous Alloys Commodity Committee, J. C. Mertz, chairman of the Corrosion and Heat Resistant Alloys, and R. C. Heath, chairman of the Rocket and Missile Panel.

In appointing the group, Division Chairman N. E. Promisel stated that the AMS organizational study is timely in view of the new, unusual, and sometimes exotic materials being used and planned for in aero-space industries.

SEAT BELT INSTALLATION — A guide to the safe and adequate installation of seat belts in motor vehicles is now available as Technical Report 177, "Motor Vehicle Seat Belt Installation." Developed by the SAE Motor Vehicle Seat Belt Committee, it was created to assure that the adequacy of seat belts provided by industry is not affected by improper installation.

A-12 SCOPE BROADENED — Committee A-12, Aero-Space Vehicle Alighting and Coupling Systems (formerly Aircraft Landing Gear Shock and Control Mechanisms), has broadened its scope to include the development of standards covering all aspects of contact and attachment systems for existing aircraft and those which will operate from earth to other planets or space stations. In addition, its activity will extend to mechanical systems for take-off and alighting as well as for retarding devices such as drag parachutes. N. W. Magyar of The Martin Company is chairman of this group.

G. C. NEWELL, Boeing Airplane Co., has been appointed chairman of the High Temperature Panel by A-6 Chairman B. R. Teree. Newell succeeds Frank Mittell, who died last May. Under Mittell, the Panel prepared the proposed MIL-H-8891 (ASG) which is the general specification for the design, installation, and tests for Type III (−65 to 450 F) hydraulic systems used on manned aircraft. The Panel is currently pursuing a study of brazed fittings.

ROY P. TROWBRIDGE, director of GM's Engineering Standards Section, has been appointed chairman of Sectional Committee B18, Standardization of Bolts, Nuts, Rivets, Screws and Similar Fasteners. Other SAE representatives serving on this committee are J. A. Boxall, K. G. Roth, and C. M. Wright.

A NEW FORMAT for existing SAE Handbook reports on aluminum and magnesium alloys is being devised by a special Editorial Subcommittee recently established by C. F. Nixon, chairman of the SAE Nonferrous Metals Committee. Reorganization of this material will do three things:

- Present new and existing data in a more legible form.
- Facilitate the addition of new grades of nonferrous metals as well as the deletion of grades which have become obsolete.
- Reduce the volume of current reports by nearly 50%.

Revision of reports on copper alloys, miscellaneous alloys and metals joining materials are also contemplated.

HUMAN ENGINEERING — Man-machine relationships are being probed by the new Human Engineering Subcommittee of the Construction and Industrial Machinery Technical Committee. With Elmer Kemp, GMC Euclid Division, to head it, the group will study human behavior and physiology in an effort to produce reports which will promote efficiency.

ENVIRONMENT — Work of the CIMTC Subcommittee XV on Environment (formerly Winterization) has been expanded to encompass the effects of arctic, desert, and tropical conditions on the design and operation of construction and industrial machinery and equipment. Chairman M. R. Nicholson, Allis-Chalmers Mfg. Co., reports that the Subcommittee's 4th Progress Report on Winterization will soon be ready for submission to the Military. This report details projects carried out over the last four years.

NEW SUBCOMMITTEE CHAIRMEN of the Construction and Industrial Machinery Technical Committee include: L. L. Evert, GMC's Euclid Division, Subcommittee I — Drawbars and Tractor Equipment Mountings. M. M. Coker, Caterpillar Tractor, Subcommittee IV — Hydraulic Power Controls. P. P. Polko, International Harvester, Subcommittee V — Electrical Equipment. H. M. Kennedy, Superior Equipment Co., Subcommittee XI — Tractor Mounted Side Booms.

LUBRICANT TYPES and their recommended use in components of construction and industrial machinery are described in a new SAE Information Report titled "Lubricant Types — Construction and Industrial Machinery." Its use by equipment designers will simplify servicing requirements by narrowing-down the number of lubricants now recommended for use.

This report will appear in the 1961 SAE Handbook.

CRC Military Work Up Again in 1960

RESEARCH activity carried on for the Military Services has surpassed the 50% mark, the 1960 Annual Report of the Coordinating Research Council reveals. Notable are new projects dealing with jet fuel icing, turbine lubrication, and instruments for measuring exhaust gas.

Jointly sustained by the SAE and American Petroleum Institute, CRC is currently supporting 43 projects, 19 of which were undertaken this year. Set up on an industry rather than a product basis, CRC work is guided by one of three committees. These groups direct all aviation, diesel, and motor research by

- Indicating the emphasis to be placed on various subjects.
- Controlling the speed with which projects are executed.
- Reviewing reports emanating from groups under them.

Highlights of this work are summarized below.

Aviation Projects

Electrostatic Discharges in Aircraft Fuel Systems — To obtain more data on hazards which may exist when turbine fuels are pumped through aircraft fuel systems, a test program is being pursued which will encompass:

- Rig tests (requiring 150 gallons) using existing instrumentation to study fuel characteristics, correlate present laboratory tests, determine the effects of tank geometry and hardware.
- Full-scale tests to correlate rig and laboratory tests and to develop information to assist in evaluating hazards in actual service.

Thermal Stability of Aviation Turbine Fuels — Development of a test technique and equipment to evaluate the thermal stability of turbine fuels used in existing military aircraft has led to a 2-phase program which will:

- Define the thermal breakpoint (in terms of preheater and filter deposits) of approximately five fuels in the High-Temperature Research Fuel Coker over a range of reservoir temperatures and preheater-filter temperatures.
- Characterize fuels in the High-Temperature Research Fuel Coker using the standard CFR Fuel Coker in an effort to build a modification kit which will permit thermal stability characterization of turbine fuels in the same order as in the High-Temperature Research Fuel Coker.

Effect of Contaminants of Thermal Stability of Turbine Fuels — Sample handling techniques designed to minimize storage and contamination effects on test programs are being probed to determine the magnitude of these effects as well as the possible effect tank coatings have on these fuels.

Jet Fuel Icing — A request from the Military Services has set in motion a program aimed at evaluating and/or developing techniques for studying problems related to jet fuel icing. Existing test techniques related to icing, water separation, corrosion, and material compatibility have already been reviewed.

Aircraft Engine Exhaust Gas Composition — A preliminary sampling of exhaust gas from a jet aircraft engine, operated under take-off and ground handling conditions, has been analysed in the chromatographic equipment at the Bureau of Mines. Results of this and other tests on full-scale engines confirm that identifiable hydrocarbons, oxides of nitrogen, and carbon monoxide in exhaust gases are extremely low. Further study to determine the extent of the problem is going on.

Surge Pressure Control in Fuel Systems CRC investigations extend in this area to ground and flight refueling, measurement of vapor pressure at high temperatures, measurement of specific heat in the high-temperature range, and detection of water in fuel.

Predicting Service Performance Properties of Lubricating — Miniaturization of component designs and future high-speed and high-altitude military aircraft have imposed new performance requirements on lubricants used throughout an airframe.

The high-temperature part of this work has been gradually extended from 300 to 700 F. CRC efforts in this area have yielded the following:

- Test equipment suitable for use under loaded conditions has been developed, and arrangements are being made to have two units available for cooperative evaluation.
- A modification of an air-driven gyro is being used as the test unit is evaluating grease in 35-size bearings operating at 20,000 rpm at 400 F.
- A rig has been developed to see if grease characteristics minimize fretting corrosion in antifriction bearings under conditions of thrust-loading.
- Equipment for evaluating the same characteristics of airframe greases under radially-loaded conditions has also been developed, and the reproducibility of the proposed test is being ascertained.

Bonded Solid Lubricant Coatings — A test technique which gives results that can be correlated with the lubricant and equipment performance requirements of existing and future aircraft are being sought. When finalized, it will be used in field studies of bonded solid lubricant coatings.

Gas Turbine Lubrication — To secure better coordination between aircraft engine builders, petroleum refiners, and the Military Services, test programs have been set up to improve test techniques or develop new ones in the following areas: Deposit and oil degradation characteristics, bearing and gear fatigue, gear scuffing, bench tests, and explosive limits.

For the present, work is limited to tests applicable to organic liquid lubricants under service conditions ranging in temperature to 700 F. Emphasis is being placed on temperature ranges that are more severe than the requirements of the current MIL-L-7808C specifications.

In addition, an oil deterioration test is also being developed to assist the Air Force.

Diesel Projects

Smokemeter — The perfecting of instrumentation and techniques for evaluating smoke from diesel engines in the laboratory continues. Emphasis is being laid on establishing satisfactory sampling techniques, revising directions for operating smokemeters, and establishing a common means of expressing smoke density in diesel engine exhaust.

Railroads — Before the end of 1960, a CRC survey team set up to look into railroad engine lubrication problems will have inspected the seven railroads covered in this program. Under inspection are a representative number of parts from diesel engines which are being observed for deposits, engine wear, corrosion protection to engine metals, frictional characteristics, and general characteristics pertaining to life and durability of engine parts.

Effect of Diesel Exhaust on Smoke and Odor — On cooperation with M.I.T., a fundamental study of compression ignition combustion is underway. The basic program is financed by a grant from the National Science Foundation, and the Ordnance Corps is making funds available to expedite the program which will cover a variety of test fuels.

The importance of this work is such that a CRC group is evaluating the need for a cooperative program on the exhaust gas problems as it affects the compression ignition engine. If warranted, a research program will be carried out by an outside organization with capabilities in this field.

Motor Projects

Exhaust Gas Composition — One of the most important projects under study relates to exhaust gas composition. Development and improvement of sampling and analytical techniques for determining important major constituents of exhaust gases, classes or organic compounds, and individual compounds continues.

However, a basic need still exists for a method of analysis which could be used as a standard to which other techniques could be compared. The nearest approach to an absolute standard, at present, is gas chromatography. The Bureau of Mines, under CRC contract, has developed a technique for this purpose, but the equipment involved is complex. To establish reproducibility of results obtained from gas chromatograph units used in industry, a cooperative program for the exchange of fuel samples has been initiated.

To expedite a search for suitable instruments, the Bureau of Mines is under contract to investigate instruments for continuous analysis of exhaust gas streams. The Bureau will provide facilities, organization, and technical knowledge for this evaluation program in which prototype models of proposed instruments will be tested at the Bureau's Bartlesville Petroleum Research Center, Okla. This method of monitoring the development of instrumentation will eliminate the need for industry to purchase a number of prototype instruments and carry out evaluations in individual laboratories.

Vapor Lock — Projects in this area are aimed at developing:

- A technique for establishing vapor handling characteristics of current automobiles.
- A better expression for measuring vapor lock that Reid vapor pressure at 100 F.
- A technique for determining the effect of fuels and vehicles on hot starting performance.
- Information on the vapor-liquid ratio temperature characteristics of volatile petroleum mixtures.

Fuel Knock Ratings — Antiknock characteristics of fuels in automotive vehicles resulting form single-cylinder tests, the significance of surface ignition, and the combustion-chamber deposit and user relationship to various erratic combustion phenomena are all being investigated.

Rumble Rating — A rumble rating program encompasses the study of a full-throttle acceleration technique for establishing vehicle rumble requirements and fuel rumble ratings in 1960 cars. It will also determine differences in fuel ratings measured in test cars of the same make and model.

Predicting Seal Performance — Tests for predicting seal performance in actual operation are under development. This two-phase project extends to both fuels and lubricants.

Power Transmission Fluids — The repeatability and reproducibility of a research technique for determining the oxidation resistance and thermal stability of these fluids are being checked, and work is almost completed on the development of a bench test for measuring the low-temperature viscosity of power transmission fluids. It is hoped the latter will yield results which can be correlated with full-scale transmission performance.

Coordinated Projects

Studies related to storage stability of fuels (motor, Jet, and diesel) and the evaluation of special petroleum products are being guided by the CRC Coordination Committee.

Fundamentals of storage stability continue to be studied by the CRC Advisory Group which in turn transmits the results of their study to the Ordnance Corps.

CRC

SAE at Missouri School of Mines

The Rolla giant (above) stands some seven feet tall, outside the university's mechanical engineering building, wearing his SAE emblem, to announce SAE Student Branch meetings.

SAE STUDENT BRANCH at Missouri School of Mines & Metallurgy is looking toward the future, in addition to actively participating in current activities. The students are studying their Student Branch Constitution for possible revisions and improvements in the operation of the Branch.

During the past year they revised their election system by voting to:

1. Elect officers through closed ballots mailed to each member.
2. And, to select candidates through a nominating committee.

They also voted to revise their governing system by electing four chairmen for the Program Committee. Each chairman will be assigned an area of activity chosen from fuels and lubricants, auto and diesel engines, aircraft, and parts and accessories, and will be responsible for one meeting in his assigned area.

For future activities, the Branch hopes to:

1. Have an information booth on campus during Freshman Week to inform the Freshmen of the aims and activities of SAE.
2. Operate a local economy run.
3. And, present an award to the outstanding Freshman enrolled in Mechanical Engineering, although no plans have been made as to the form of the award.

The Branch's current activities include: monthly meetings, field trips, a vehicles safety check, and two technical paper contests.

The monthly meetings include movies on technical subjects and topics of general interest, such as "The Indianapolis 500", and speakers from Detroit, Chicago, and St. Louis. Each year the Branch hears from six to eight speakers. Among the eight speakers they heard during 1959–1960 were: Kenneth J. Fleck of Caterpillar Tractor Co. who spoke on "Combustion Process in the Diesel Engine"; Phillip Rasson of the Corvette Division of the Chevrolet Assembly Plant in St. Louis who discussed "The Corvette"; E. W. Tanguary of Innational Harvester Co. who spoke on "Development of Tomorrow's Mechanical Equipment"; and, B. K. Flanery of Westinghouse Electric Corp., whose topic was "Gas Turbines and Exotic Alloys."

Door prizes, donated by neighboring business concerns, are often given at the Branch meetings. After the meetings, refreshments are served in the Mechanical Engineering Laboratory and the students are given an opportunity to meet the speaker personally.

One or two field trips are taken each year to an industrial plant in the St.

Aerial view of the campus at Missouri School of Mines & Metallurgy.

Louis area. This past year the students visited the new Chrysler Assembly plant in St. Louis.

A vehicle safety check, which is sponsored by the local Kiwanis Club, is operated each year by the SAE Enrolled Students. The local officials, who are very cooperative, restrict a two block area along side the campus for the check. The event lasts for one week and approximately 1500 cars are checked.

A technical paper's contest is sponsored by the Branch at their February meeting each year. This contest has a two-fold purpose: to encourage students to write and present technical papers, and to pick three students to represent the Student Branch at the St. Louis Section Paper's Contest.

St. Louis Section's March meeting is reserved for Students Night, and, for the last six years, has been a technical paper's contest between the Student Branches at Parks College and Missouri School of Mines. The students compete for prizes of $50.00, $30.00, and $20.00, and each student that participates receives his first years dues in SAE from the St. Louis Section. The winning school, on total points, receives a banner to display until the next contest. Since the banner was created, each school has won it three times. The Missouri Student Branch won this past year. Two Missouri seniors Douglas Kline and John Wake won first and second place, respectively.

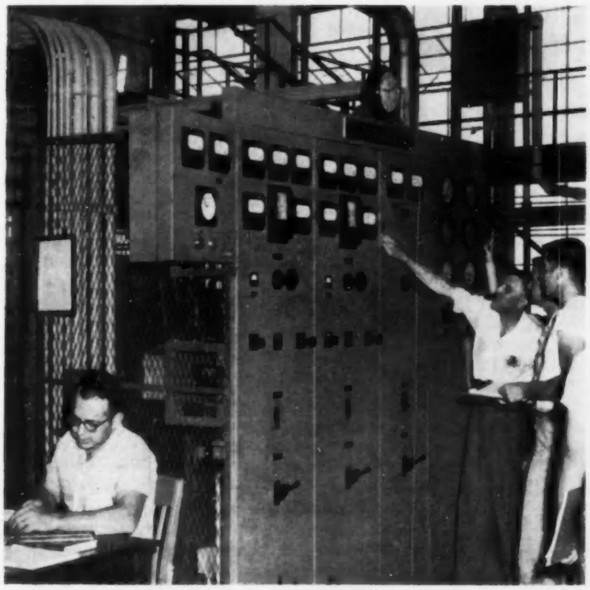
The Missouri School of M & M SAE Student Branch received its charter in 1953, under the encouragement of C. R. Remington, Jr., associate professor of mechanical engineering, who is an active member of St. Louis Section. The success of the Branch, which is one of the largest Student Branches in SAE with 175 members, was due primarily to the efforts of Prof. Remington and G. L. Scofield, professor of mechanical engineering. The Branch's current activity and progress is encouraged by the leadership of James A. Jones, who is their Faculty Advisor.

JAMES A. JONES, assistant professor of mechanical engineering, is Faculty Advisor to the SAE Student Branch at Missouri School of Mines & Metallurgy.

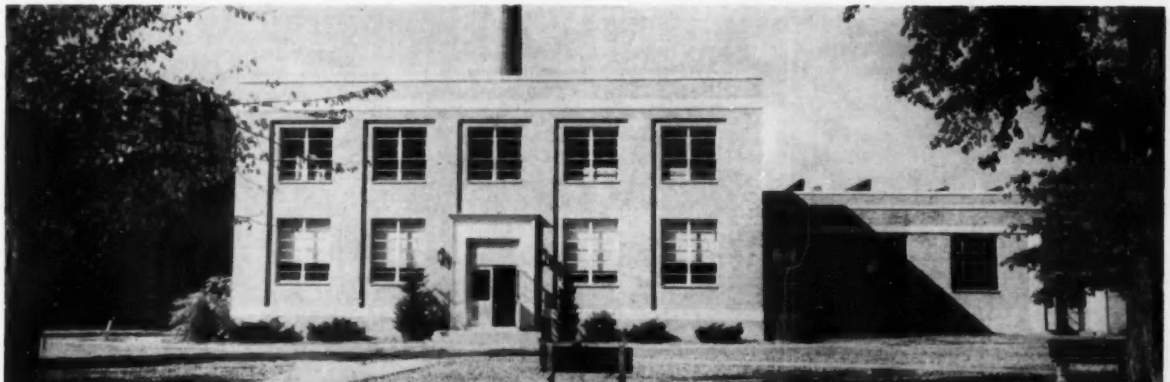
A native of Centralia, Ill., Jones graduated from the Missouri School of Mines in 1956 and received his master's degree from there in 1960. For three years he was an instructor of mechanical engineering at the University and has been assistant professor since 1959.

In addition to being Faculty Advisor, Jones is serving SAE as 1960–1961 vice-chairman of student activities for the St. Louis Section.

Jones



The steam power equipment (at left) is a part of the facilities found in the school's mechanical engineering laboratory.



The university's mechanical engineering building.

SAE Members

L. W. CHRISTENSON has retired as vice-president in charge of sales for Cleveland Graphite Bronze Division of Clevite Corp.

Christenson joined Cleveland Graphite in 1924 and developed the company's sales in the Chicago area. He became sales manager in 1943 and was elected vice-president in 1948.

In retirement Christenson will engage in sales and marketing consulting.

W. D. COWGILL will succeed Christenson as head of Cleveland Graphite's sales organization. Cowgill was named general sales manager four months ago after serving for eight years as the company's Detroit sales manager.

WALTER H. SHEALOR has been appointed assistant director of sales for International Divisions of Timken Roller Bearing Co. He had been assistant general manager of the company's Export Division.

WALLACE T. MILLER has retired as chairman of the board of directors at NTH Products, Inc.

An alumnus of University of Michigan Law School, Miller was one of the Founders of NTH in 1938. He has also been chief accountant of Willis Overland Co., treasurer and a director of Mutual Motors Co., and general sales manager of Motor Wheel Corp. During World War II, he directed defense operations at Studebaker's Pacific Coast plant.

Miller plans to spend his time as sales representative for Calcor Corp., Whittear, Calif. and United Pattern & Foundry Co., Los Angeles.

C. R. BOLL has been appointed to the new position of executive vice-president. marketing at Cummins Engine Co., Inc. Boll has served the company as sales engineer, assistant regional manager, manager of engine sales, general sales manager, and, most recently, vice-president for sales.


Christenson


Shealor


Miller


Boll


Ault


Larson


Abbott


St. George

THOMAS J. AULT has been named president of the Automotive Division of Budd Co.

Ault joins Budd after 23 years with Borg-Warner Corp., where he served in various executive capacities. Resigning from Borg-Warner in 1958, he has since served as president and director of Saco-Lowell Shops, Boston.

NORMAN LARSON has been appointed manager of original equipment sales for Thompson Products Piston Ring Division of Thompson-Ramo-Wooldridge, Inc. Previously he was district manager of Piston Ring Division's Milwaukee office.

RALPH G. ABBOTT has been appointed chief engineer for Ensign products at American Bosch Division of American Bosch Arma Corp.

Abbott was previously division engineer for Ensign Carburetor Co., which was acquired by American Bosch Arma Corp. a year ago, and has been merged with the corporation.

A. JOHN ST. GEORGE has been transferred to American Bosch Arma Corp.'s newly created Commercial Sales Division, and will serve as Ensign product manager. He had been sales manager at Ensign Carburetor Co.

GEORGE A. HACH has been appointed sales manager of non-ferrous metals for Central Foundry Division of General Motors Corp. Previously he served the division as product development engineer.

EARL L. MONSON has retired as assistant chief engineer at American Motors Corp.'s Kenosha Division.

Monson joined the company at Milwaukee in 1920 as experimental mechanic. In 1925 he became experimental engineer. He was transferred to Kenosha plant in 1936 as a quality engineer and during World War II served as special test engineer in the firm's aircraft engine program. In 1945 he was named chief development engineer, and in 1955 he was appointed assistant chief engineer.

CARL A. LOY has been named aeronautical research engineer for National Aeronautics and Space Administration at the George C. Marshall Space Flight Center at Huntsville, Ala.

Loy had been senior structural engineer for Missiles & Space Systems Division, United Aircraft Corp.

N. HEATH McDOWELL has become salesman for B. Gray, Inc. He had been general manager, Pacsal Division, Illinois Tool Works. McDowell has been in the general management field for 15 years and has always found sales interesting.

JAMES N. SOWERS has become director of methods and standards for American Airlines, Inc.

Sowers has served Ford Motor Co. for the past ten years and was most recently assistant manager of quality control at their Wixom Plant, Wixom, Mich.

GILBERT WAY, of Ethyl Corp., moved his office to Los Angeles when the company closed its San Bernardino Laboratories. His office address is now 1141 Huntley Drive, Los Angeles 26, Calif. His home address is 505 N. Ivescrest Ave., Covina, Calif.

LOUIS J. STANKIEWICZ has become a research specialist for Rocketdyne, Division of North American Aviation, Inc. He had been engineering manager at Cleveland Graphite Bronze.

DAVID J. O'LEARY is now district representative for Florida and Southern Georgia, Industrial Division, J. I. Case Co. He had been product specialist at Massey-Ferguson, Inc.

WILLIAM B. PASHKOW has been appointed president of WARD LaFrance Truck Corp. For the past five years, Pashkow has been associated with Mack Trucks, Inc. He was most recently manager of government product development at Mack Trucks.

WILLIAM HARVEY JOLLIFF, previously associate engineer, is now design liaison engineer for Convair Astronautics in San Diego.

FRANK HENRY ABAR, JR. has become supervisor of instrumentation section at Ford Motor Co., Transmission & Chassis Division. Formerly he was product engineer at the Ford Division.

United Aircraft Changes

Parkins

Martin

WRIGHT A. PARKINS retired September 30 as vice-president for engineering and a member of the board at United Aircraft Corp. He will continue to serve the corporation as a consultant.

A graduate of the University of Washington, Parkins joined Pratt & Whitney in 1928. He is widely known for his work in the growth and development of Pratt & Whitney Aircraft engines, both piston and gas turbine, and particularly for his contributions to the reliability of the modern aircraft powerplant. He was directly responsible for the development of Pratt & Whitney Aircraft's postwar piston engines and supervised the engineers and technicians who brought through the division's current turbine engines and rocket powerplants.

ERLE MARTIN has been named to the new position of vice-president for research and development at United Aircraft Corp. A vice-president since 1952 and a member of the board since 1958, Martin will assume the engineering, research, and development responsibilities for the corporation. He has been directing the operations of both the Hamilton Standard and Norden Divisions for the past two years.

K. H. HOFFMAN, former manager of transmissions operations of Allison Division, General Motors Corp., has been named administrative assistant to the general manager.

R. E. LYNCH, former manager of aeroproducts operations of Allison Division, General Motors Corp., has been named to succeed **K. H. Hoffman** as manager, transmissions operations.

TED BAYLER has joined Atlantic Research Corp. as manager of reliability and quality assurance. Formerly he was quality control reliability coordinator for Chrysler Corp.'s Missile Division.

ROBERT W. PODLESAK has been appointed manager of Chevrolet's 21 manufacturing plants for General Motors Corp. He had been regional plant manager in charge of Chevrolet's manufacturing plants at Flint, Saginaw, and Bay City, Mich.

GEORGE KAPPELT has been appointed director of engineering laboratories for Bell Aerosystems Co. He had been chief metallurgist at Bell since 1948.

Kappelt is chairman of Non-ferrous Alloys Committee of SAE Aero-Space Materials Division.

LEO A. PFANKUCH, formerly vice-president and general manager of Pacific Scientific Co., is now president of Shur-Lok Marine Corp.

ROY B. BENDER has been appointed vice-president of Automotive Division of Studebaker-Packard Corp. He will retain his present position as Parts and Service Division manager in addition to his new capacity.

WILLIAM L. MOSHER, JR. has been appointed regional plant manager of Chevrolet assembly plants in western U. S. for General Motors Corp. He had been manager of Chevrolet manufacturing plants in Detroit, Buffalo, Tonowanda, N. Y., and Massena, N. Y.

— continued —

SAE Father and Son

John Twells

Robert Twells

JOHN LAWRENCE TWELLS is shown with his father, **ROBERT TWELLS**, an SAE member since 1945.

The younger Twells a University of Toledo graduate, joined The Electric Autolite Co. as a special representative based in Atlanta, Ga., after two years service in the U. S. Army.

Robert Twells is vice-president and group executive of the Ceramic Division and Spark Plug Division of Electric Autolite Co. He is currently a member of the SAE Engineering Materials Activity Committee.

E. W. SMITH, formerly sales manager, is now manager of Heat Transfer Products Division of Yates American Machine Co.

BERNARD J. LEHMAN has joined Delco-Remy Division of General Motors Corp. as engineer in training. He had been vice-president for Glacier Peat Moss Corp.

RICHARD L. COURTNEY is now automotive engineer for the N. Y. district, Ethyl Corp. Formerly he served the company as project and research engineer.

CHARLES W. WILHIDE has become chief specifications engineer for Perfect Circle Corp. Previously he served Goodyear Aircraft Corp. as senior development engineer.

HARRY A. NELSON has joined Litton Systems, Inc. as senior engineer. Previously he was project engineer for United Aircraft Products, Inc.

ROBERT W. SPINNER has been named director of reliability at General Motors Corp. He has been staff engineer in charge of new car testing at Buick since 1958.

DONALD LLOYD HAINWORTH, formerly production planning engineer, is now manufacturing engineer for Westinghouse Electric Corp.

JOHN H. VANDERBILT has joined Hamilton Standard Division of United Aircraft Corp. as designer for ground support equipment. Formerly he was test engineer at Stratos Division of Fairchild Engine & Airplane Corp.

JACK W. FRENCH is now division manager of research and engineering at Lone Star Boat Co. He had been department manager for Lincoln Advanced vehicles at Ford Motor Co.

WALTER J. OLIVER has joined L. E. Myers Co. as mechanical engineer. Previously he was layout engineer for Automatic Transportation Co.

N. SURYANARAYANA MURTHY, formerly a student, is now lecturer in mechanical engineering at Indian Institute of Technology at West Bengal, India.

J. ARNOLD KIELY has become assistant to the president for Cone Automatic Machine Co., Inc. Formerly he was vice-president at Saco Lowell Shops, Auto Division.

DUANE H. GOODSMITH is now sales engineer at Kaiser Aluminum & Chemical Sales, Inc. He had been account manager.

CLIFFORD E. WILLIS, previously vice-president and general manager for Cleveland Pneumatic Industries, has established his own business as a management consultant.

ROBERT W. FLEMING, formerly senior engineer for Westinghouse Electric Corp., is now staff engineer for General Nuclear Engineering Corp.

CHARLES B. GALE has joined Standard Oil Co. of Indiana as supervisor of sales engineering schools and technical bulletins. Formerly he was employed by American Oil Co. as manager of administrative services.

DONALD E. McMINN has become president of Eagle Mfg. Corp. He had been sales manager of Coldform Division of H. K. Porter Co.

J. FRANK CANFIELD, of White Motor Co., is taking part in a series of special training courses at the company's home office in Cleveland.

Obituaries

RALPH L. ELLINGER . . . (M'44) . . . western representative and head of equipment and planning development for Trans World Airlines, Inc. . . . died September 11 . . . born 1896.

DAVID T. EVANS . . . (M'39) . . . retired . . . died August 5 . . . born 1891.

FRANK K. FRASER . . . (M'45) . . . plant manager and member of board of directors, Permafuse Corp. . . . died August 27 . . . born 1900.

DENIS C. GASKIN . . . (M'43) . . . vice-president and general manager of Mack Trucks of Canada, Ltd. . . . died September 10 . . . born 1898.

ALBERT R. JACOBS . . . (M'41) . . . retired vice-president and founder of Jacobs Aircraft Engine Co. . . . died September 2 . . . born 1894 . . . was named Girard "Man of the Year" in 1945.

A. JAMES SAMWAYS . . . (M'54) . . . chief engineer, Toro Mfg. Corp. . . . died August 24 . . . born 1908.

HERBERT H. SCHULDT . . . (M'40) . . . administrative assistant to vice president of engineering and research at Schwitzer Corp. . . . died May 10 . . . born 1915.

FRANK A. SHARPE . . . (M'25) . . . retired . . . died July 14 . . . born 1883.

WILLIAM F. STERNBERG . . . (M'58) . . . western regional manager of Diamond T Motor Truck Co. . . . died May 5 . . . born 1912.

Rambling . . .

Through The Sections

The NEW UNISEAL OIL RING which is being tested extensively by Ramsey Corp., is essentially a one-piece flexible circumferential type oil ring. Testing of the ring was of particular interest to members of **ST. LOUIS SECTION** who toured the Ramsey plant and observed actual engine testing in their dynamometer laboratory, September 13.

The self-locking Spirolox retaining ring with its ability to solve high rotational speed and/or impact loading applications was also of interest. It was discussed and considered a possible answer to industry's need for a retaining ring of this type.

MORE AIRFIELDS are needed to accommodate the growing number of utility aircraft, W. T. Piper of Piper Aircraft Corp. told **WILLIAMSPORT GROUP** September 12.

Elaborate high cost airports are not necessary, he said, but simple airfields of the "Grass-Strip" variety which are easily accessible to small towns and every aircraft owner.

THERE IS NO PRODUCTION FACILITY available at present which is large enough to produce all the exotic alloys that would be required if gas turbines were to go into automotive type mass production in their present state, I. A. Swatman, of Ford Motor Co., told **INDIANA SECTION** September 15.

The Ford 704 300 hp gas turbine engine concept, however, has the potential of equaling or surpassing a diesel engine of equivalent horsepower in fuel consumption, weight and reliability, Swatman stated.

A NUCLEAR APPROACH to making a 1250-KW nuclear reactor-steam turbine power plant suitable for transportation to, set-up, and operation in remote sites which is being developed by the Martin Co. was described by Ronald G. Dugas of the company's Nuclear Division at **BALTIMORE SECTION** September 8.

EXTREME ACCURACY needed in the instrumentation and control of a space guidance system is illustrated by the fact that an error in speed of only one foot per sec would result in an error of approximately one mile from a target on the surface of earth. The same discrepancy would result in an error of 20–100 miles from a target on the moon and of 40,000 miles on Venus, Dr. Walter T. Olsen of National Aeronautics and space Administration, told **MID-MICHIGAN SECTION** September 26.

IN 1911 Clyde Cessna designed, built, and learned to fly his first flying machine, Gerald Deneau, of Cessna Aircraft Co., told **WICHITA SECTION** on September 15 as he reported the development of Cessna aircraft from the early pre-Airmaster through 1960 models.

WITHOUT COMPACTS fuel usage would have increased 5–6% in 1960; whereas it actually increased only 3–4%, Robert W. Hogan of Ethyl Corp. told **ATLANTA SECTION** September 7.

In 1957, 4.1% of all passenger car sales were compact cars (including imports). Today (1960) 32% are compact sales, and in 1961 40% are estimated to be compact sales.

ONLY SEVEN OR EIGHT TIRE SIZES were required for original equipment manufacturers ten years ago; whereas today eighteen are required, M. P. Hershey of Firestone Tire & Rubber Co. told **SOUTH BEND DIVISION, CHICAGO SECTION** September 19.

Today's annual rubber consumption is 4,000,000 tons. The projected requirements for 1970 are 7,000,000 tons, I. J. Sjothun of Firestone told the Section. The difference, he said, must be "made up" in increased production of synthetic rubbers.

Today 25% of passenger car tires use nylon cord, 75% use tyrex, an improved rayon, B. M. Wolf also of Firestone stated. Nylon is used exclusively for high performance racing tires, and truck tires double the nylon percentage. Dacron is a very promising "miracle" fiber for future cord use.

Seated at the September 19 meeting of CHICAGO SECTION, SOUTH BEND DIVISION are: (left to right) B. M. Wolf, manager of textiles and adhesives, Firestone Tire & Rubber Co.; M. P. Hershey, manager passenger and airplane tire engineering for Firestone; M. P. DeBlumenthal of Studebaker,Packard Corp.; and I. J. Sjothun, senior project chemical engineer for Firestone.

MILWAUKEE SECTION CHAIRMAN G. A. Rea (center) was on hand to welcome D. G. Thomas, general chairman of the Meeting (left) and M. L. Frey, chairman of the Production Forum Planning Committee (right). Milwaukee Section acted as host for the Meeting.

PARTICIPATING AT THE MEETING were the Powerplant Activity Committee and the Farm, Construction, and Industrial Machinery Activity Committee. Gregory Flynn, Jr. (left) and W. F. Shurts (right) are the respective chairmen of these committees.

PRODUCTION PROBLEMS OF SMALL ENGINES were discussed by distinguished panel of experts. Left to right, J. G. Boehm, Eric Bonow, Leo Stoll, William Hoth, P. E. Speiser, and G. J. Bell.

PRECISION TOOLING FOR JOB SHOP OPERATION was discussed by knowledgeable panel consisting of (left-to-right) Nick Beresnoff, Harry Johnson, Jim Shultz, D. E. McHenry, W. D. Clark, and F. A. Cuthbertson.

FCIM Meeting Stresses New Developments

Advances in design, engineering, and production of farm, construction and industrial machinery cited at Milwaukee.

RECENT DEVELOPMENTS in design, engineering, and production were spotlighted at the 1960 SAE Farm, Construction, and Industrial Machinery Meeting in Milwaukee.

The broadened technical program featured presentations on boom design, clutches and brakes, corrugated skins for earthmovers, air cleaner efficiency, theory of steering, structural analysis, and component life investigation of agricultural equipment. Papers were also presented on steel properties, case crushing of gears, hydraulic filters, new tractors, styling, powertrain bearing life, elastomeric vulcanizers, fuel and lubricant additives, and aircooled and water-cooled diesel engines.

The Production Forum consisted of panel discussions on heat-treatment, engineering-manufacturing relationships, unique manufacturing processes, welding methods and process control, production control tools for scheduling and shop loading, gear and spline design and production, operating experience with numerically controlled machines, precision tooling for job shop operation, and the production problems of small engines.

Recent Developments

Nine truck and bus fleets using a multipurpose automotive grease containing molybdenum disulfide as an additive reported that tests over a two-year period for a total of two million miles showed a 50% or more reduction in parts wear.

The field tests also indicated that "moly" grease, when used for chassis lubrication, greatly improved ease of steering and performance of "fifth" wheels. In actual applications the new grease resulted in fewer parts replacements, lower labor costs, and less frequent need for lubrication.

The reason for molybdenum disulfide's excellent properties as an additive lies in its molecular structure, which consists of layers of molybdenum atoms sandwiched between layers of sulfur atoms. The sulfur layers slip past each other easily and account for the low friction in a grease containing "moly."

Also described at the meeting was an infrequently used, infra-red spectroscopy test which reveals what happens to an additive in gear lubricants during service. The laboratory test enables researchers to obtain basic information about the kind of additive needed to meet the exacting and often contradicting requirements of gear lubricants.

A proposal was made at the meeting that the tractor industry cooperate to gather actual operating data of tractors in service on farms, so that studies could be made of the fatigue life of powertrain components in relation to the power used by the tractors. This could lead to better and more economical tractor designs.

Sixty-five per cent of the farm tractors and ninety per cent of the construction machinery in Germany use aircooled engines, SAE'rs were told. And, the trend is to continuously increased production of aircooled diesel engines in many other European countries as well.

The popularity of aircooled engines is due to their superior reliability over water-cooled engines under the difficult conditions encountered in the agriculture and earthmoving industries.

A prediction was made that the robustness and simple maintenance requirements of the aircooled diesel will appeal to the underdeveloped countries; and that particularly in the medium horsepower range, the development of the aircooled engine in Germany and other Western European countries will promote its use in the U. S.

A type of gear failure occuring on carburized and hardened gear tooth profiles has been distinguished from pitting fatigue. Called "case crushing," the failure is characterized by cracks that extend through the hard case and deep into the soft core material. Pitting fatigue cracks, on the other hand, are usually shallow.

Another characteristic of case crushing is its sudden appearance, often without any warning. Pitting fatigue occurs gradually. A case crushing failure can be identified by a longitudinal gouge shape, quite different from the v-shape of a pitting failure. Also case crushing usually occurs on only one or two teeth, with surface cracks normal to the surface; while pitting fatigue failure occurs on many teeth at the same time, and the surface cracks run at an acute angle to the surface.

It is believed that the case crushing failures are not due to maximum shear, but are related to the ratio of shear stress to shear strength of the material below the level where the hardened case meets the softer core. Initial experimental evidence points to a critical value of 0.55 for this ratio.

A 1952 survey of large acreage farmers showed a potential market for a large four-wheel-drive tractor with 150 drawbar horsepower; so John Deere designed the 8010 tractor.

Deere engineers described the design and development problems encountered during the eight years which culminated in the introduction of the new tractor this year. The huge ten-ton tractor, which is less than 96 in. wide, underwent extensive engineering analysis in the design stage and actual stress performance evaluation in the field because up to this time Deere's experience had been primarily with smaller agricultural tractors.

The present trend in all industries where brakes are used is an increase in the severity of braking. This is due to greater energy conversion and reduction in the size of brakes. Alloyed brake drums and heavy-duty linings operating in a good flow of forced air will undoubtedly help alleviate the problem; but, if problems related to high temperatures continue to exist, the use of disc brakes, either aircooled or oilcooled, may be needed. Multiple-disc clutches and brakes have been used with success on crawler tractors,

Continued on p. 116

FCIM Meeting

HONORED AT THE TRACTOR SHOW were Future Farmers of America members (left-to-right) Roger Galford, Paul Uphoff, Jim Thomas, Glen Engelking, and Merlin Arnold. With the farm youths is Bob Dimberg who acted as M. C. for the agricultural equipment display. Thomas is president of the FFA and as such heads the 9000 chapters and 380,000 members of the organization.

ON HAND TO INTRODUCE THE FFA MEMBERS and to present timely comments was Boyd S. Oberlink, senior vice president and group executive, Allis Chalmers Mfg. Co.

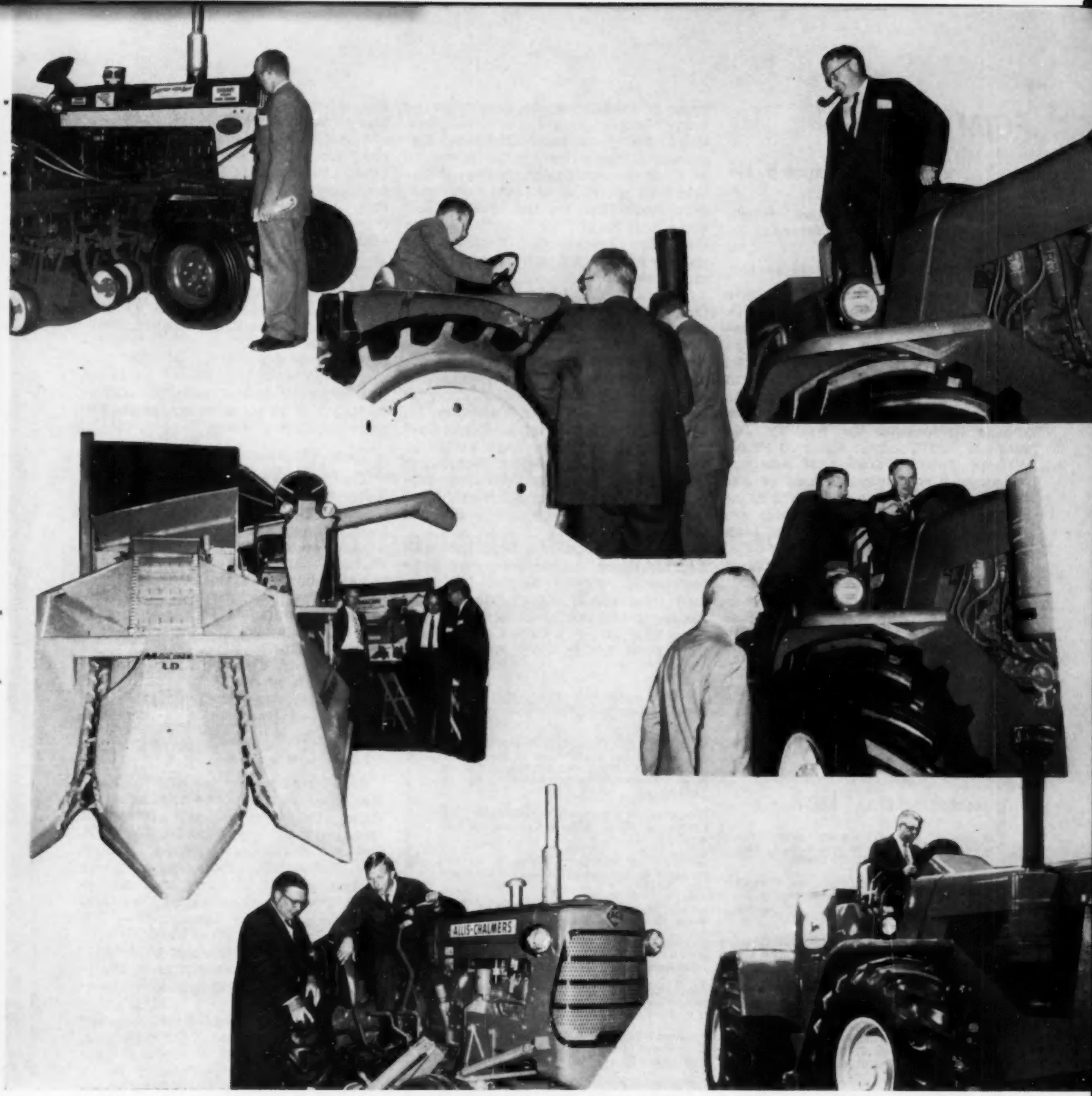
at the Tractor Show . . .

SAE PAST-PRESIDENTS William K. Creson and Ralph R. Teetor attended the Tractor Show Luncheon.

TRACTOR SHOW LUNCHEON SPEAKER Tom Anderson (left) discusses political issues affecting agriculture with Ted Swansen, vice president, Ladish Co. Anderson, president and publisher of Farm and Ranch Magazine, spoke on Problems of Government in Business. Swansen was toastmaster at the luncheon.

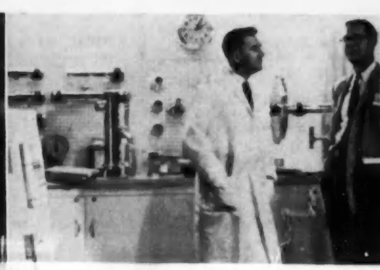
GENERAL COMMITTEE MEMBERS D. G. Thomas (right) and W. W. Henning discuss Tractor Show arrangements. Thomas was General Chairman of the Meeting; Henning acted as chairman of the Tractor Show.

MUSIC WAS SUPPLIED by the Allis Chalmers Band and Glee Club.



SAE MEMBERS GIVE NEW FARM EQUIPMENT A WORKOUT . . . Climbing aboard, looking under, peering into the latest in agricultural equipment, SAE'rs were free to satisfy their curiosities at the Tractor Show.

. . . and engineering display

FCIM Meeting

. . . continued from p. 113

and it may be only a matter of time until their use is more widespread.

Farm Youth Urges Improvements

Tractors of the future should have air conditioning, be more comfortable, have better fuel consumption, and feature more hydraulic controls, SAE's tractor engineers were told by a young farmer at the Tractor Show held in conjunction with the Meeting. As spokesman for more than 380,000 farm youths throughout the country, 20-year-old Jim Thomas, National President of Future Farmers of America, echoed the equipment desires of our future farmers. The FFA is a non-profit organization of boys who are studying agriculture in vocational high schools throughout the country.

On hand to introduce the farm youths and present timely comments was Boyd S. Oberlink, senior vice-president and group executive, Allis-Chalmers Mfg. Co. In answer to Thomas, Oberlink declared the challenge of youth must be recognized by today's tractor designers and engineers. Bob Dimberg, manager, Industrial Engineering, Allen-Bradley Co. served as M. C. of the equipment display. Dimberg also served as chairman, Attendance Promotion, for the Meeting and was a member of the Production Forum Planning Committee. Oberlink was sponsor of the Production Forum Planning Committee.

With Jim Thomas were four young farmers, regional winners of the FFA Farm Mechanics Award, who are skilled in farm mechanics and have shown considerable ingenuity in designing and constructing equipment needed on their home farms. The four award winners are Glen Engelking, 18, of Ridgefield, Washington; Roger Galford, 17, of Dunmore, West Viriginia; Merlin Arnold, 18, Welch, Oklahoma; and Paul Uphoff, 17, of Graymont, Illinois. Each youth described his achievements in farm mechanics to the box lunch gathering.

Another special event of the SAE Tractor Show was an extensive non-commercial display of the latest in farm tractors and implements. Participating in the display were: Allis-Chalmers Mfg. Co.; J. I. Case Co.; Deere & Co.; Ford Motor Co.; International Harvester Co.; Massey-Ferguson Ltd.; and Minneapolis-Moline Co.

Allis-Chalmers' orchestra and glee club serenaded the picnic luncheon group attending the Tractor Show with a number of musical selections. Backdrop for the performance was the mammoth farm machinery display in Milwaukee Arena.

Allis-Chalmers Field Trip

Two busloads of SAE'rs attended a plant tour of Allis-Chalmers in Milwaukee held as part of the FCIM Meeting schedule. Attendees had the opportunity to see tractor assemby, the machining and assembling of steam and hydraulic turbines, and the manufacturing of powerplant equipment for nuclear power stations.

Farm Magazine Publisher Speaks

The outspoken President and Publisher of Farm and Ranch Magazine, Thomas J. Anderson, addressed SAE members at a mid-week luncheon held as part of the FCIM Meeting in Milwaukee. Anderson, who is noted throughout the farm states for his hard-hitting "straight talks" on political and economic issues affecting agriculture, spoke on Problems of Government in Business. His column "Straight Talk" appears in more than 375 newspapers each month.

Anderson was introduced by Toastmaster Ted Swansen, vice-president, Ladish Co.

• • •

SAE Engineering Activity Committees participating in the meeting were: Farm, Construction and Industrial Machinery Activity Committee — W. F. Shurts, chairman — George Reuter, vice-chairman; Production Activity Committee — C. W. Ohly, chairman — K. O. Tech, vice-chairman; Fuels and Lubricants Activity Committee — Gilbert Way, chairman — E. H. Scott, vice-chairman; Powerplant Activity Committee — Gregory Flynn, Jr., chairman — C. E. Habermann, vice-chairman.

The General Committee for the Meeting was composed of D. G. Thomas, general chairman; N. P. Mollinger, reception committee chairman; R. A. Dimberg, attendance promotion chairman; R. G. Jensen, publicity chairman; Quentin O'Sullivan, public relations advisor; W. W. Henning, Tractor Show chairman; and W. C. Arnold, M. L. Frey, S. K. Rudorf, Harlan Banister, Igor Kamlukin, W. F. Shurts, J. K. Dickey, Leo Lechtenberg, F. P. Steiner, Gregory Flynn, Jr., E. H. Panthofer, Gilbert Way, R. C. Frevik, and George Reuter.

Serving on the Production Forum Planning Committee were: Boyd S. Oberlink, sponsor; M. L. Frey, chairman; and E. C. Bannan, R. M. Bruesewitz, R. A. Dimberg, N. P. Mollinger, R. C. Mortensen, Don Monson, J. A. Nelson, S. K. Rudorf, Leonard Shore, Giles Smith, and R. D. Teece.

Papers . . .

presented at FCIM Meeting

The Use of Corrugations in Construction Equipment, **H. V. Parsley** and **L. L. Lemke,** International Harvestor Co. **(216A)**

Some Factors Affecting Dry Air Cleaner Efficiency, **E. H. Farnan,** Ford Motor Co. and **J. A. Weber,** University of Illinois **(216B)**

Detergent Additives for Gasoline and Diesel Engine Lubricants, **C. V. Smalheer,** The Lubrizol Corp. **(217A)**

Paradoxical Products — Lubricants for Gears, **E. L. Will,** Monsanto Chemical Co. **(217B)**

Molybdenum Disulfide as an Additive to Improve the Performance of an Automotive Multipurpose Grease, **C. D. Thayer** and **H. G. Rudolph, Jr.,** Socony Mobil Oil Co., Inc. **(217C)**

Vehicle Steering Fundamentals, **W. H. Baier,** Armour Research Foundation **(218A)**

The Advantages of Structural Analysis in the Design Stage of Agricultural Equipment, **M. A. Walck,** Ford Motor Co. **(219A)**

Power and Life Investigation of the Farm Tractor Drive Components, **A. J. Jenkins,** Timken Roller Bearing Co. **(219B)**

Fatigue, Impact, and Tensile Properties of High Strength Leaded and Non-Leaded SAE 4140 Steel, **William Simon,** Westinghouse Electric Co. **(220A)**

"Case Crushing" of Carburized and Hardened Gears, **R. Pedersen** and **S. L. Rice,** Caterpillar Tractor Co. **(220B)**

Gasoline Additives — A Review for Engineers, **A. E. Felt** and **H. C. Sumner,** Ethyl Corp. **(221A)**

Additives for Liquid Hydrocarbon Fuels, **H. E. Collins** and **E. C. Squerciati,** Du Pont Petroleum Lab. **(221B)**

Air-Cooling vs. Water-Cooling in Agricultural and Construction Equipment, **H. H. Haas,** Klockner-Humboldt-Deutz A. G., **(222A)**

Principles of Replaceable Absorbent Filter Cartridges, **H. A. Wilson,** Commercial Filters Corp. **(223A)**

Surface Type Hydraulic Filters, **H. L. Forman** and **C. J. Casaleggi,** Purolator Products, Inc. **(223B)**

Mobile Hydraulic Systems Filtration, **L. E. Terry,** Sperry Rand Corp. **(223C)**

The Potential of Structural Mechanics Research in Cran Boom Design, **D. B. Singer,** Lockheed Aircraft Corp. **(224A)**

Excavator Clutch and Brakes, Basic Properties of Friction Materials, **A. J. Bette,** Johns-Manville Corp. **(224B)**

Excavator Clutches and Brakes, Behavior of Drums and Linings, **R. W. Lange,** American Brake Shoe Co. **(224C)**

Engineering Deere's Model 8010 Tractor, **M. L. Miller, F. C. Walters** and **Robert Tweedy,** John Deere Tractor Research and Engineering Center **(225A)**

John Deere's New Line of Tractors, **Merlin Hansen,** John Deere Tractor Research & Engineering Center **(225B)**

The Stylist: The Engineer-Stylist Relationship, **R. Ten Eyck,** Richard Ten Eyck Associates **(226A)**

The Engineer: The Engineer-Stylist Relationship, **R. Adee,** Hesston Mfg. Co. **(226B)**

Styling and Functional Aspects of Utilizing Reinforced Polyester Resins in Fertilizer Distribution Equipment, **M. C. Christensen,** Ford Motor Co. **(226C)**

Bearing Life Analysis for Torque Converter Driven Power Trains, **H. B. Scheifele,** Federal-Mogul-Bower Bearing, Inc. **(227A)**

Papers are available through SAE General Publication Department. Prices: 50¢ a copy to members; 75¢ a copy to nonmembers.

COBO HALL • DETROIT • JANUARY 9-13 • 1961

INTERNATIONAL S·A·E AUTOMOTIVE ENGINEERING CONGRESS · EXPOSITION

Preview of **TECHNICAL SESSIONS** at the

1961 SAE International Congress and Exposition of Automotive Engineering

Over 175 papers will be presented on the following subjects – don't miss any of these important developments in your field . . .

Monday, January 9

9:00 A.M.

Shortening the Time Cycle for Automobile Body Tooling

Short cuts to automotive design and tooling plus new ways to make forming tools result in faster body changes at lower cost.

Effects of Weight and Size Limitations Changes Proposed by AASHO

A review of how the new size and weight code proposed by American Association of State Highway Officials will effect the design, development, and operation of trucks and the design of highways and bridges.

9:00 A.M. AND 1:15 P.M.

Future Short-Haul Transportation: Ground vs. Air, Part I and II

How will the passenger travel in 50–400 mile journeys in mass transportation in 1970 — monorail, buses, people pods lifted by crane helicopter, STOL craft . . . ?

Seating and Interior Dimensioning of Passenger Cars, Part I and II

Researchers from General Motors, Harvard University, France, and Japan bring you basic information on these very important subjects.

Things You Ought to Know about Batteries — Part I and II

Batteries for electric vehicles, physical chemistry of lead oxides, design, service life, testing, and other fundamentals.

Gas Turbine Compressors, Part I and II

Design of axial, centrifugal, and mixed-flow compressors for small gas turbine powerplants for aircraft, ground vehicle, and stationary power applications.

3:30 P.M.

Ground-Effect Machines

Advantages of various GEM cycles. Status of GEMs under development for military and civilian service over land and water.

Effect of a Fuel Cell Powerplant on Future Automobile Body Design

Will development and introduction of fuel cells or some equally radical powerplant result during the next decade in sweeping changes in passenger car body design?

Compression Temperature Measurements in Diesel Engines

Research on compression temperatures and their effect on ignition presented by researchers from Penn State and University of Wisconsin; use of water-vapor radiation and

continued on next page

SAE International Congress and Exposition of Automotive Engineering

Monday, January 9 (continued)

fine-wire thermocouples for measuring temperatures; influence of rpm on peak temperature and startability.

An Improved Method of Specifying Rubber Parts for Automotive Use

Explanation of the new plan for systematizing and simplifying designations for all the elastomeric vulcanizates.

The XM521 Experimental Research Vehicle

Being developed under the U. S. Army Ordnance Advanced Design Program for Wheeled Vehicles, the XM521 experimental 2½-ton truck incorporates bonded body-frame structure, unusual swimming and driving characteristics, very light-weight design, compact size, low silhouette, and one-to-one payload-to-weight ratio.

Tuesday, January 10

9:00 A.M.

Effects of Wet and Icy Runways and How to Clear Them

NASA and Swedish studies of effect of wet and icy runways on jet transport take-off and landing performance, plus descriptions of latest snow-removal equipment developed by the Canadian Department of Transport.

A Breakthrough for Bodies — Glass, Polymers and Plastics

As the properties, availability, and competitive position of upholstery and trim materials, glass and other nonmetals change during the next 10 years how will they affect passenger car styling?

Alternators and Storage Batteries for Small Engines

Voltage requirements for small engines. The small alternator and its advantages over d-c generators.

Generating New Ideas on Production Processes

Ability of engineers to generate new ideas, either singly or in groups, can be cultivated. What approach is best — "brainstorming", "pyramiding", "rugged individualism" or . . . ?

Wire Cord Tires

Construction, working principles, service life, and recapping characteristics of wire cord tires developed in Europe and America.

Thermodynamics and Combustion

British survey of advances in design of combustive chambers for spark-ignition and compression-ignition engines . . . and report on influence of cylinder-bore size on detonation.

9:00 A.M. AND 2:30 P.M.

Developments in Heat Exchanging for Aerospace Vehicles and Reactors — Part I and II

Advances in the art of cooling (1) nozzles for liquid rockets, (2) cast turbine blades, (3) supersonic airframes, (4) re-entry capsules and other aerospace vehicles. Also news of molten metal loops for absorbing heat of nuclear reactors.

12:00 noon—Luncheon Honoring SAE's Overseas Guests

2:30 P.M.

Passenger and Cargo Handling Equipment

Chrysler's mobile lounge for carrying passengers from terminal to jet transport; military needs for cargo handling equipment; Canadian and European systems for loading and unloading.

Corrosion-Proof All-Metal Bodies

Since many body members have also become structural members (in unitized bodies), they need better protection against corrosion. How to choose appropriate corrosion-resistant metals — and insure success in forming, drawing, welding, and finishing them.

Automobile Electrical System Developments

A new alternator, electrical-system requirements for city driving, radio interference, and design of systems using diodes and transistors.

Designing Trucks for Super Highways

How will the increasing number of high-speed limited-access highways affect the design of trucks? What are the aerodynamic considerations Can drive train parts be smaller and less complicated? How will the fleet operator benefit from specially designed vehicles?

Laboratory Knock-Testing Methods of Improved Significance

Changes in engines and fuels require new knock-testing methods. New techniques for determining knocking resistance of fuels including over-100-octane-number fuels will be discussed.

Standard Units of Measurement

A proposed gradual program for introduction of the metric system. The results of a General Motors survey of what such a change would mean. A review of the educational implications of the two systems.

Practical Application of Operations Research

How can Operations Research programs be held within practical limits and what is the pay-off when this is accomplished?

5:00 P.M. {Student Engineers Night / Fathers and Sons Night}

8:00 P.M.

The Rotary Combustion Engine

Performance data on the NSU-WANKEL type of rotating combusting engine. Comparison with reciprocating engines and comment on possible applications.

Wednesday, January 11

9:00 A.M.

New Uses of Isotopes in Research

German studies of friction phenomena. Other research applications of radioactive materials.

New Powerplants for Trucks and Trains

Ford's 704 gas turbine, and the hydraulic-drive locomotive of General Motors of Canada.

Metallurgical Investigation and Accelerated Testing

Tests conducted in Spain lead to new system for interchangeability and classification of steels. Accelerated testing at the FIAT laboratories.

Coordinating Engineering and Manufacturing Effort for Improved Product Reliability

How can all of today's experience in reliability, including military experience, be adapted to fit the problems of competitive industry?

Automatic Transmissions — 1961

Engineering features of several automatic transmissions developed for new compact and standard passenger cars.

9:00 A.M. AND 1:15 P.M.

Abnormal Combustion — Part I and II

Abnormal combustion behavior of gasolines; combustion phenomena at engine cranking speeds; new insights into the mechanism of surface ignition; and maesurement of flame temperature by sodium-line reversal method.

1:15 P.M.

Useful Effects of Ionizing Radiation

Petroleum products, plastics, coal, and other organic materials can be made more useful by subjecting them to charged particles from accelerators or other radiation sources.

Multifuel Engine

New data from the German firm of M.A.N. on multifuel engine practice.

Automobile Suspension Systems

Fundamental data and theory on the elasticity characteristics and other dynamic properties of the modern automobile body structure.

1:15 AND 3:30 P.M.

Reliable Electrons — Part I and II

Truck electrical system reliability . . . a joint review by the users, vehicle manufacturers, and suppliers.

3:30 P.M.

Fluid Gearing — The L. Ray Buckendale Lecture

Design of single-stage, three-element torque converters including determination of torus diameter, blading design, performance estimation, fabrication, effect of fluid density and viscosity, engine matching, altitude performance, and cooling.

Breakthroughs in Engineering Education

Educators and industry leaders will review (1) trend toward regrouping courses of study, (2) engineering education in foreign lands, (3) on-the-job re-training of engineers, and (4) industry's desires in the college training of engineers.

Military Needs for Small Turbines

Army, Navy, and Air Force views on future applications for up-to-300-hp gas turbine engines and the consequent performance requirements.

Securing Maximum Flexibility of Production Equipment

Standard machine tools can be linked together to create highly productive, but not too specialized, production lines for moderate-volume production.

6:30 P.M. MAIN BANQUET HALL
—COBO HALL

Annual Dinner

PRINCIPAL SPEAKER . . .

John F. Gordon

President, General Motors Corp.

Thursday, January 12

9:00 A.M.

Motor Oil Development Criteria

Low-temperature cranking and flow properties of waxy polymer-thickened motor oils. Development of anti-wear properties in motor oils. The CLR engine as a tool for evaluating sludging tendencies of motor oils.

Breakthroughs in Computers

New developments in computer units and new applications of computers to automotive engineering problems.

Better Passenger Car Transportation at Less Cost

Long-range views about the man-machine-environment system, reducing the cost of car ownership, and safety on the highway.

Excavating by Nuclear Blasts

The power of the atom may offer a new earthmoving technique for use under water. Applications in other large difficult areas are also being considered.

Transistorized Ignition

Review of the past year's experience with transistorized ignition systems in fleet operations and comparisons with conventional ignition systems.

9:00A.M., 1:15 AND 3:30 P.M.

"Project Moonbeam" — a 10,000 1-lb-Payload Lunar Vehicle — Part I, II and III

Propulsion, navigation, guidance, communication, landing, life-support, and re-entry provisions for a hypothetical vehicle to make regular trips to and from the moon.

continued on next page

Preview of TECHNICAL SESSIONS at the

SAE International Congress and Exposition of Automotive Engineering

Thursday, January 12 (continued)

1:15 P.M.

Meeting Future Mobility Needs of the Motorist

To meet our future mobility problems, we must plan far ahead. What can be done now to meet our future requirements?

Disc Brakes and Anti-Skid Braking Devices

Experts from England, Germany, and the United States will review current activity in the design and development of these units.

1:15 AND 3:30 P.M.

Fuel Filter Test Methods — Part I and II

Report of new testing techniques in evaluating fuel filter capacity, pressure drop and volume flow — and evalution of media migration.

3:30 P.M.

Test Tracks and Engine Air Cleaners

Reports on the British Ministry of Supply's all-weather test track and sarch for the perfect engine air cleaner.

Containerization of Freight

A state-of-the-art report on shipping freight in specially designed containers revealing how truck fleet operators, airlines, railroads, and steamship companies are affected.

3:00 AND 8:00 P.M.

New Engines for the 1961 Passenger Cars — Part I and II

New passenger-car engines, ranging from the first U. S. die-cast aluminum passenger-car engine to a 4-cyl powerplant that was originally developed as an 8-cyl engine will be discussed. New GM aluminum V-8 engines as well as Studebaker's new six described in detail. Men who made the engineering decisions on these engines will outline their reasoning.

Friday, January 13

9:00 A.M.

Theory of Vehicular Traffic Flow

Developments in the theory of traffic flow, including the follow-the-leader model of single-lane traffic flow, deductions from studies of vehicular flow in "tunnel" situations, and use of statistical models.

Bus Developments, Operations, and Maintenance

A review of the developments in American and European buses since World War II and a study of future trends predicted from consideration of road designs and traffic regulations; competition from passenger cars, trains, and airlines; and the effect of maintenance experience and unique operating situations.

Fuel Cells and Small Reactors

Fuel cells for powering ground vehicles; compact reactors that can be moved from site to site.

Road-Vehicle Loading Relationships

German and American studies of the loads imposed by the wheel on the road, and by the road on the vehicle. Tips to designers on how to design vehicles that will cause less road wear.

9:00 A.M. AND 2:00 P.M.

Transmission Workshop — Part I and II

Continuation of the highly successful Workshop Series — this time covering design of transmission oil pumps and friction elements.

2:00 P.M.

Magnetohydrodynamics

A review of the basic principles of MHD . . . and imaginative expositions of how MHD may be applied in space propulsion, ground vehicle propulsion, and for clutches, brakes, and other automotive applications.

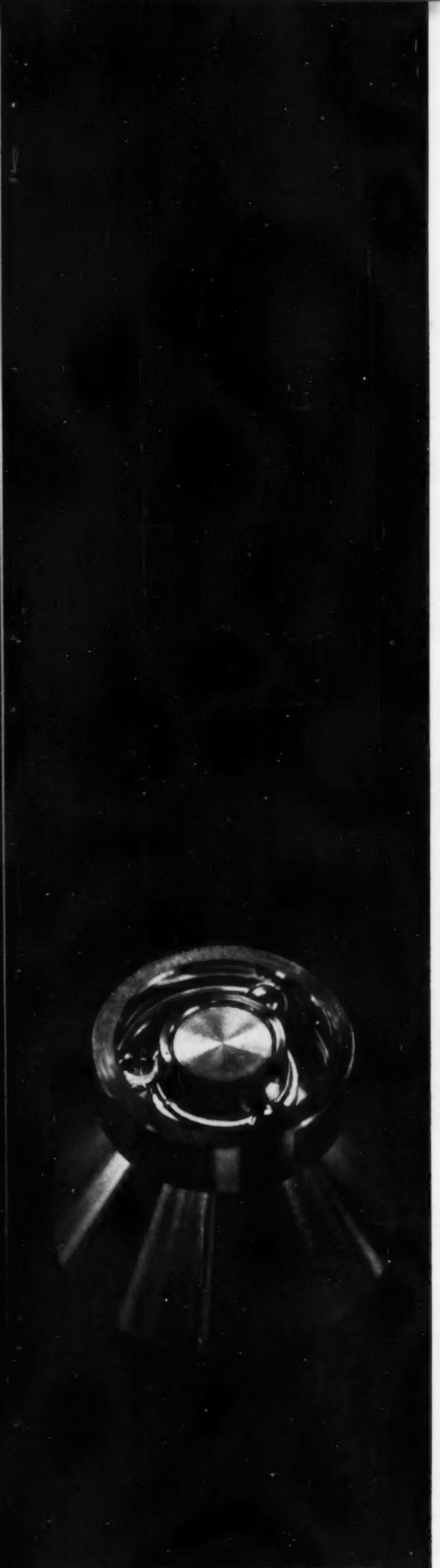
Compact Trucks for Utility Use

The new compact trucks being introduced by American manufacturers will fill the need of many city fleets. Papers will describe several of these vehicles, including construction details, and outline how utilities can adapt these units.

Over 175 papers will be presented at TECHNICAL SESSIONS at the

1961 SAE International Congress and Exposition of Automotive Engineering

COBO HALL • DETROIT • MICHIGAN • JANUARY 9-13 • 1961



One of a series

Catching the Drift of Gyro Bearings

The fantastic accuracies needed by inertial guidance systems for space flight depend on the suppression of gyro drift, the tendency of a gyro to precess from minutely occurring internal torques. Particularly puzzling has been the problem of "jogs," or sudden axial shifts, within gyro spin-axis bearings. Shifts of but one ten-millionth of an inch can cause serious steering error.

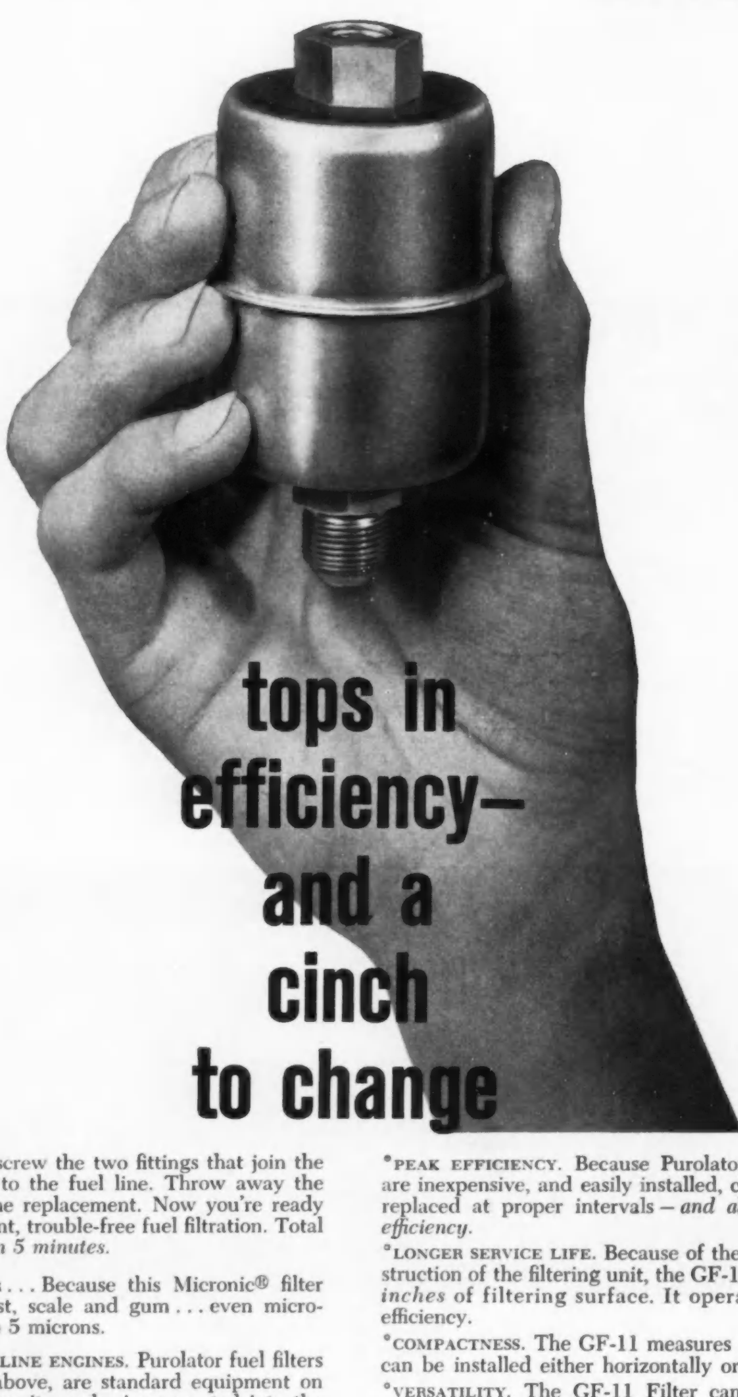
Specialists at the GM Research Laboratories have found that the real key to drift lies in the thickness and distribution patterns of bearing lubricating films. Only a tenth of a milligram of oil — equivalent in volume to less than two-thousandths of a drop of water — is required in a gyro bearing, but even this amount unevenly distributed may cause jogs.

Conducting unique studies of single bearings apart from rotor assemblies, GM Researchers use a hydrostatic spindle and special instrumentation to take film-thickness measurements they compare with hydrodynamic theory. Jogs, due to excess oil supply, have been analyzed in relation to surface oil transfer and separator feed control, ball spin orientation, displacement, and differential heating and ball wander.

This experimental and analytical approach is achieving progress toward jog-free, stably distributed, and suitably thick oil films required in high-precision bearings. It is a further example of the critical and advanced research General Motors carries out in seeking "more and better things for more people."

General Motors Research Laboratories
Warren, Michigan

The fluoresced streaks show the disturbed "wake" of the lubricating film during bearing operation. The active part of the film, too thin to fluoresce visibly, averages ten-millionths of an inch in thickness.



HERE'S ALL YOU DO! Unscrew the two fittings that join the Purolator GF-11 Filter to the fuel line. Throw away the dirty filter. Fasten in the replacement. Now you're ready for 5,000 miles of efficient, trouble-free fuel filtration. Total time involved? *Less than 5 minutes.*

MOTORS RUN SMOOTHER . . . Because this Micronic® filter removes dirt, metal, rust, scale and gum . . . even microscopic particles down to 5 microns.

ADAPTABLE TO ALL GASOLINE ENGINES. Purolator fuel filters like the GF-11 shown above, are standard equipment on most 1960 cars. However, it can be incorporated into the fuel system of almost *any* gasoline engine — automotive, portable or marine.

MORE ADVANTAGES. Easy installation and replacement is a big reason for the popularity of the Purolator GF-11 Fuel Filter. But here are more:

°PEAK EFFICIENCY. Because Purolator replacement filters are inexpensive, and easily installed, chances are they'll be replaced at proper intervals — *and always work at peak efficiency.*

°LONGER SERVICE LIFE. Because of the special pleated construction of the filtering unit, the GF-11 has fully *70 square inches* of filtering surface. It operates longer at peak efficiency.

°COMPACTNESS. The GF-11 measures about 3″ by 1⅞″. It can be installed either horizontally or vertically.

°VERSATILITY. The GF-11 Filter can be installed as an O. E. M. item on practically any gasoline-engine — from sport cars and garden tractors to power mowers and midget racers.

For complete information on the GF-11 and other Purolator filters, write to Purolator Products, Inc., Department 3849, Rahway, New Jersey.

Filtration for Every Known Fluid

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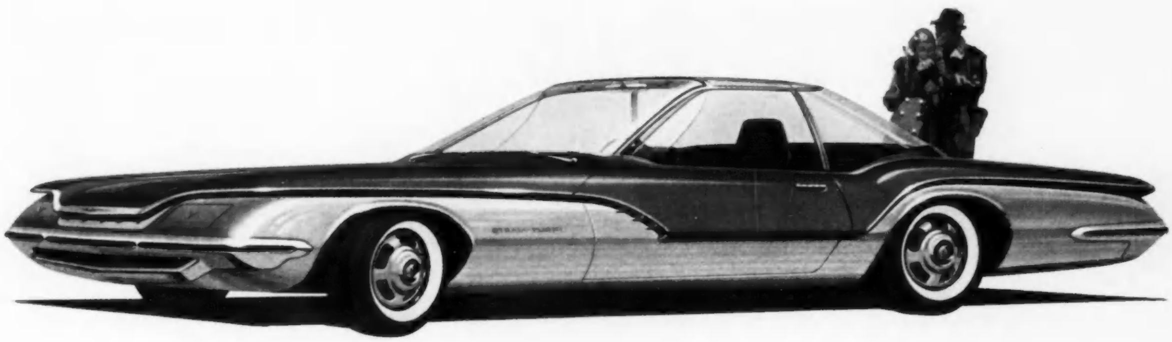
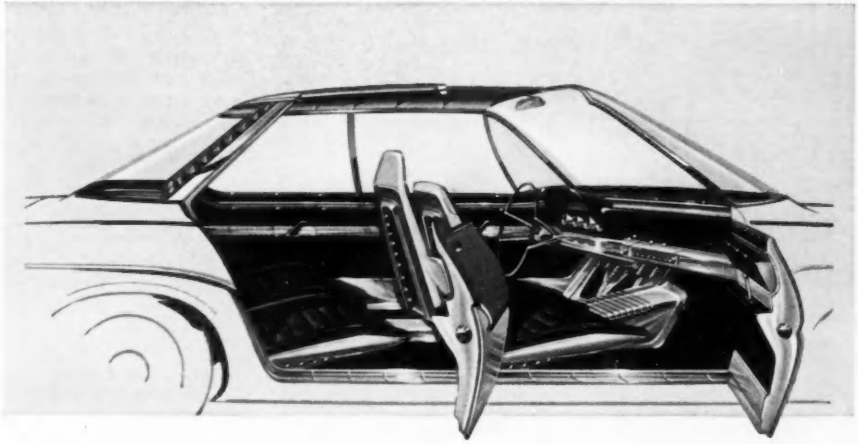
Rahway, New Jersey, and Toronto, Ontario, Canada



stainless steel

No other metal has the strength, beauty and versatile qualities that serve you so well today and promise so much for tomorrow.

There is nothing like stainless steel for AUTOMOBILES



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Manufacturers of high quality Stainless and Carbon Steels

Stainless Steel

Look for the **STEELMARK** on the products you buy.

McLOUTH STAINLESS STEEL

McLOUTH STAINLESS STEEL

Briefs of SAE PAPERS

continued from page 6

Approaches to Direct Flame Afterburner, G. E. CORNELIUS. Paper No. 210B. Research done at Holley Carburetor Co., and development of multiple venturi type afterburners for reduction of truck and bus engine air pollution; results of tests showing its limitations; design and construction of muffler replacement type afterburner; study of three ways of controlling excessive temperatures during deceleration cycle, consistent with satisfactory hydrocarbon reductions; criteria to consider in designing pilot burners; models of muffler replacement type afterburners.

Need for New Concept of Rapid Transit, N. KENNEDY, W. S. HOMBURGER. Paper No. 210C. Influence of transportation on past and present growth of American cities and reasons why present concepts of rapid transit do not seem to be suited to urban growth, to serve present and future urban populations; study being conducted by Chicago Transit Authority on feasibility of system that allows manual operation of vehicles as single units in low density suburban residential areas and automatic operation, possibly in trains, in high-density central cities.

Directional Stability and Control of Four-Wheeled Vehicle in Flat Turn, M. GOLAND, F. JINDRA. Paper No. 211A. Analysis takes into account variation of cornering performance of tires with changes in their vertical loading, investigating steady, flat turn to appraise effects of turn radius and forward speed on steering characteristics and dynamic directional stability; results of calculations are presented for vehicle series in which fore and aft location of center of gravity is varied.

Full Scale Testing of Highway Bridge Rails and Median Barriers, F. N. HVEEM, J. L. BEATON. Paper No. 211B. Tests were made of 15 designs of traffic barriers for use in highway medians to develop efficient barriers for use in freeway construction; new barrier design was developed and several installations are now on California highways; report presents technical findings of collisions with different classes of highway traffic barriers, rigid, semi-rigid, and flexible; procedures and data reduction methods employed.

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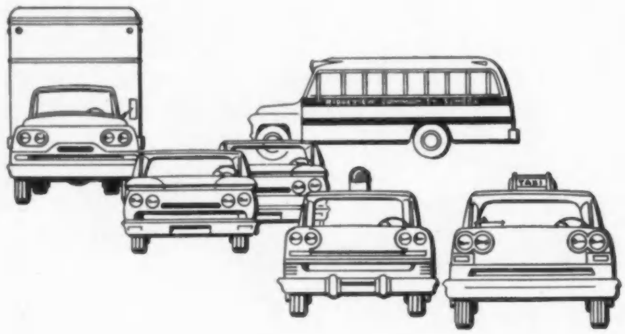
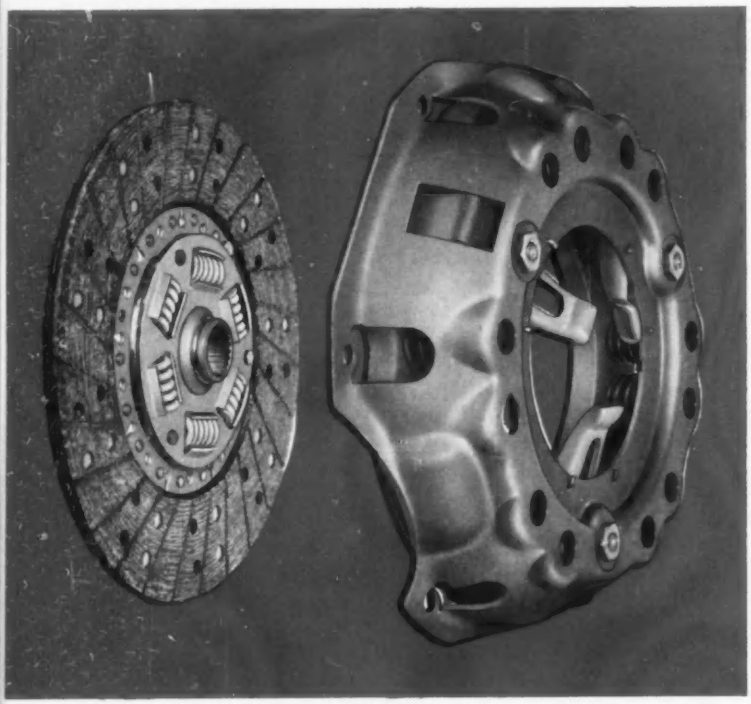
with turbine not operating; additional

Design and Muskeg Operation of 20



New Interchangeable BORG & BECK Clutches for Fleets, Police Cars, Taxis

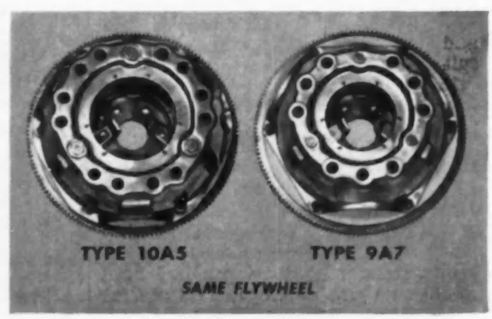


More Capacity—no increase in bolt circle or flywheel size

Now there's no need to change the bell housing, flywheel, motor mounts or pedal linkage when converting cars for fleet, police, taxicab or other heavy duty service. Borg & Beck's new A5 clutches are designed specifically for these installations, as well as for trucks and school buses—provide the additional capacity required, yet are interchangeable with the next smaller size of Borg & Beck Types A7, A8 or A9 clutches.

Type 10A5, for example, mounts on the same flywheel bolt circle as Type 9A7—yet is rated at 265 ft.-lbs. torque capacity compared with 210 ft.-lbs. for the 9A7.

Like all Borg & Beck clutches, the new Type A5 clutches are designed, engineered and built to Borg & Beck's leadership standards for quality, performance and value. That's your assurance of complete satisfaction. Consult our engineers for full details.



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Briefs of SAE PAPERS

continued from p. 127

Crash Studies of Modern Cars with Unitized Structure, R. H. FREDERICKS, R. W. CONNOR. Paper No. 211C. Collision characteristics of modern cars; crash performance of 1958 Lincoln and 1950 Falcon were compared to behavior observed with 1955 Mercury when all were crashed into rigid barrier; unitized structures are comparable with frame body construction in terms of potential protection in collision when occupants avail themselves of safety features and devices developed.

Head-on Collisions, Series III, J. H. MATHEWSON, D. M. SEVERY, A. W. SIEGEL. Paper No. 211D. Two fully instrumented automobile head-on collision experiments, equipped with anthropometric dummies, were carried out by Univ. of Calif. for impact speeds of 23 and 32 mph; data include car and motorist deceleration patterns for many locations, injury analyses and evaluation of relation that vehicle collapse characteristics bear to motorist force and injury patterns; equipment, facilities, instrumentation.

Diesel-Gas Turbine Power, C. G. A. ROSEN. Paper No. 212A. Utilization of diesel turbocharging to expanded advantages is explored to reveal flexibility of combinations and diversity of possibilities of diesel engine-gas turbine combination; elements of military vehicle powerplant installation; concept which satisfies these requirements proposes diesel-gas turbine power plant; advantages of 2-shaft, free-floating gas turbine and diesel engine are shown; schematics; starting cycle.

Electrochemical Fuel Cell for Transportation, M. EISENBERG. Paper No. 212B. Desirability of employing fuel cells in electric automobiles; advantages of electric transportation; fundamental principles of electrochemical fuel cells; use of redox cells; it is found that as far as ground transportation in automotive applications is concerned, chemical fuel cells operating from inexpensive fuels are indicated approach; future research needed.

Gas Turbine Cycles and Design Concepts for Vehicle Propulsion, C. H. PAUL, E. L. KUMM. Paper No. 212C. Report shows how load and its characteristics can affect gas turbine design, and proposes concept of dual powerplant whereby gas turbine is used to supply starting power for diesel engine; diesel can be used over its lower power range in normally aspirated fashion, with turbine not operating; additional power demands can be met by both clutching in turbine to drive shaft and using turbine to supply pressurized air to diesel; turbine bleed air can be used for air conditioning or heating.

Designing Off-Highway Mobility, W. M. BROWN, V. DORSEY. Paper No. 213A. Approach used by Kenworth Motor Truck Co. in designing trucks for use in canefields of Hawaii, muskeg of Alaska, forests of Philippines, deserts of North Africa or Middle East; requirements and development of 1-p high deflection sand tire, picking axles, drive lines, transmissions, clutches, and engines for intended service; data on three models used in various tire combinations; chart on tire data, tread type, make and revolutions per mile.

Design and Muskeg Operation of 20 Ton Payload Carrier, Musk-Ox, C. J. NUTTALL, Jr., J. G. THOMSON. Paper No. 213B. Muskeg, or organic terrain, is most difficult transportation barrier in northern Canada, singularly unfriendly to overload vehicles; 25% of prospective oil land in Canada is muskeg covered; review of existing articulated tracked vehicles, and development of Musk-Ox tracked carrier designed to transport wildcat drilling rigs for imperial oil; carrier is powered with Cummins NRTO-6 turbocharged 6-cyl diesel engine; overall performance; field experience.

Electric Transmission for Railroad Locomotives, T. B. DILWORTH. Paper No. 214A. General characteristics of components of electric transmission

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Briefs of SAE PAPERS

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used in diesel electric locomotives; description of each component, how these components are combined and controlled in electric transmission in EMD General Motors locomotives and results which may be obtained from system.

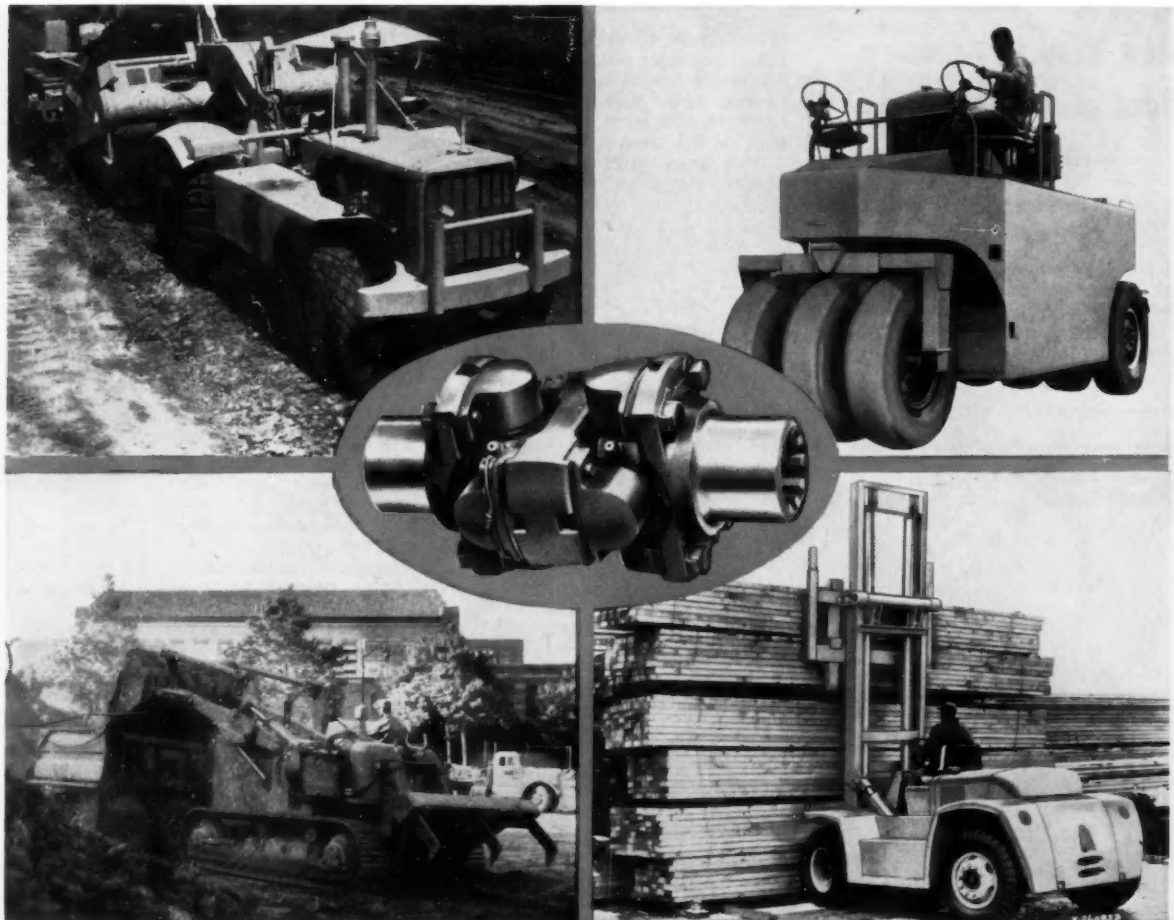
Development of Small Automotive Diesel in Western Europe and Its likely Role in U. S. A., J. H. PITCHFORD. Paper No. 215B. History for development and design programs in various countries; economics of diesel from user's and national viewpoint; shortcomings of small compression ignition engines and measures being taken or to be taken to overcome these features; for small utility size vehicle rapid extension of use of diesel engine is foreseen.

MATERIALS

Mating Materials in Unlubricated, High-Load, Low-Speed Wear Tests at High Temperature in Air, L. E. FULLER. Paper No. T-43. Report on tests designed to represent service conditions of high-temperature valves, control linkage bearings, pneumatic cylinders and seals; equipment, procedures, methods of evaluation, and results from tests run at speed of 10 fpm under contact load of 500 psi at 1000 and 1400 F; of combinations tested chromium carbide coating (1C-IA) vs aluminum oxide coating (LA-2) was best and 1C-IA mating with itself second.

Performance of Springs at Temperatures Above 900 F, W. R. JOHNSON, R. D. CROOKS. Paper No. T-44. Details of material evaluation test using modified wire torsion tester, designed to test helical compression springs for use in jet engines, rockets, aircraft wheel brakes, etc; Inconel X was used as basic experimental material; data for relaxation and creep are presented on performance of statically loaded spring materials for temperatures up to 1200 F and times up to 1500 hr.

Effect of Material and Processing Variables on Residual Stresses in Carburized and Induction-Hardened Gears, E. T. BERGQUIST, S. M. LENHOFF. Paper No. T-45. Study correlates residual stresses with existing heat treating practices for steels employed in tank and other tracked vehicle final drive assemblies; over 300 pinions of different materials were processed; results show that optimum residual compressive stress (S_t value equal 42,700 psi compression) is attained by gas carburizing, slow cooling, reheating, and brine quenching 8620 steel.



New Way to Adhesive Bond at 3000-6000 F

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is first carburized to tungsten monocarbide by a gas carburizing process. . . . Then, the graphite is vapor- or electro-plated with tantalum, which is then gas-carburized to tantalum monocarbide.

The two carbides are then placed intimately together in vacuum or atmosphere at 4600 to 4800 F . . . . and held in contact until the solid solution formation is affected. A catalytic compound, such as CoO may be used either to accelerate the solution formation or the formation of an eutectic.

This same technique can thus be adapted for joining together a variety of high-melting-point base materials, whereby the method does not put a joint limitation on the assembly.

■ **To Order Paper No. 233A . . .** from which material for this article was drawn, see p. 6.

Material and Processing Influence Gear Stresses

Based on paper by

E. T. BERGQUIST
Western Gear Corp.

and

S. M. LENHOFF
Ordnance Tank — Automotive Command
U. S. Army Ordnance Arsenal Detroit

RESIDUAL TENSILE STRESSES are thought to shorten the fatigue life of carburized and induction-hardened gears. Residual compressive stresses, on the other hand, tend to offset operationally opposed surface tensile stresses and thus prolong service life. If these assumptions are true, the optimum processing technique for a given material would result in a relatively high amount of residual compressive stress.

An investigation was begun in 1956 to correlate residual stresses with existing industrial heat-treating practices for various commercial steels for the ultimate improvement of tank and other tracked vehicle final-drive assemblies by using optimum processes. After selecting the areas for investigation the specific purposes were to:

1. Compare residual stresses resulting from the different combinations of materials and heat-treating methods.

2. Discover the combination of material and heat-treat method resulting in the most favorable residual stress pattern.

3. Determine whether the parting-out stress (S_t) correction factor is a major determining factor of surface stress. (Initial strain-gage readings were taken after heat treatment and dimensional inspection to establish "zero conditions" on the intact gear. Subsequent strain-gage readings were taken both after parting of the tooth from the sample pinion, and also after specimen sizing. The difference between the zero reading and the specimen-sizing reading was denoted as S_t. This value included two factors: (1) change in length, and (2) change in curvature.)

In this investigation, over 300 sample pinions of differen materials were processed. The results of the analysis showed that optimum residual compressive stress (S_t value = 42,700 psi compression) was attained by gas-carburizing, slow-cooling, reheating, and brine-quenching SAE 8620 steel. However, the S_t value obtained with this steel and processing method was not particularly outstanding when compared with some of the other samples treated. Considering the program as a

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KNOW YOUR ALLOY STEELS . . .

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whole, the maximum difference between the highest and the lowest S_t value was approximately 42,000 psi. It is obvious that the relationship and understanding of the role of residual stresses to gearing can be considered as an art rather than a science. The benefits of further understanding and control of these stresses are certainly worthy of the efforts which will be necessary to achieve them.

■ **To Order Paper No. T-45 . . .** from which material for this article was drawn, see p. 6.

New Flight Instruments Wait Better Reliability

Based on paper by

G. T. GEBHARDT

Boeing Airplane Co.

NEWER and more accurate instruments, and instrument systems, are needed for the jet transports currently in use, but they will not become available until the state-of-the-art of instrument making reaches a higher level. Even long-in-use, standard instruments often have low reliability and insufficient accuracy. Imperfections still existing in such devices as altimeters, air-speed and temperature-measuring instruments, impede the development of newer instruments.

An example of the need for new instruments is the lack of sufficient instrumentation for that portion of the flight regime involving the transition from ground roll to climb-out. Data given to the pilot do not tell him the amount of airplane rotation and this he must know to execute the optimum climb-out path.

The airplane attitude indicator does indicate rotation, but since it is acceleration-sensitive in pitch it is not accurate during this period of the flight. The other flight instruments such as air-speed, rate of climb and altimeter do not supply this information either. The pilot must depend to a large degree on skill and "feel" to rotate the aircraft properly. Although minor mishandling can be tolerated without detrimental effects, an under-rotation tends to consume more runway than normally required and an over-rotation could result in tail skid contact with the runway surface.

One instrument that has been considered for indicating the amount of rotation is the lift-coefficient transducer, but unfortunately the many problems and expense of making this instrument into operational production units preclude its present-day use. The

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Transverse Properties of Steel

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instrument consists of a vane located near the leading edge of the wing. The vane is deflected in proportion to the lift coefficient and dynamic pressure but the resulting signal may be modified, by a pure dynamic pressure signal, to give a needle deflection proportional to lift coefficient alone.

Tests to evaluate the usefulness of this instrument were performed in a flight simulator. Pilots with considerable jet experience were found to perform equally well with conventional instruments as with the lift coefficient device. Pilots having little or no jet experience were materially aided by the new instrument, but only when it was used in conjunction with an accelerometer. The reliability and maintenance problems associated with the accelerometer and its stabilizing gyro are the factors hampering the use of this instrument as a pitch axis flight director.

The lift coefficient device, without the accelerometer, is being currently tested in conjunction with a stick-shaker, to provide the aircraft with over-rotation and stall warning instrumentation.

Progress is being held up for lack of sufficiently accurate production line instruments, also, in air data system centralization. The trend here has been to employ servo-driven, rather than pneumatic, instruments. It has been found that servos, in some cases, are not capable of the accuracy of pneumatic instruments. In other cases, even though a servo-driven instrument can be built to attain sufficient accuracy, the instruments coming off the production line do not reach these attainable levels.

The altimeter is an example of the case where the servo type of instrument is not capable of the accuracy that a pure pneumatic type can produce. The altimeters on most of the American built jet transports are corrected by servo computers. It has been found that below 7000 ft the uncorrected altimeter is more accurate than the corrected altimeter because the computer tolerance exceeds the required correction at low altitudes.

It is felt that servo-driven instruments should be used, but they should only supplement the pneumatic instruments until experience has shown them to be reliable and of value.

No simple solution exists to the problem of reliability. It will be solved only by keeping the instrument systems as few and as simple as possible. Better design principles, quality control, and installation practices along with improved methods for maintenance, overhaul, packaging, and handling must be exercised.

■ **To Order Paper No. 247A . . .**
from which material for this article was drawn, see p. 6.

Transverse Properties of Steel Highly Significant in Gear Design

Based on paper by

WILLIAM SIMON

Westinghouse Electric Corp.

THE TRANSVERSE fatigue, impact, and tensile strength properties of high strength, leaded and non-leaded SAE 4140 alloy steel are considerably lower than the corresponding longitudinal properties. Hence gears, pinions, and similar parts which are loaded in the transverse direction should be designed on the basis of these transverse properties.

The investigation, which led to this recommendation, also revealed that

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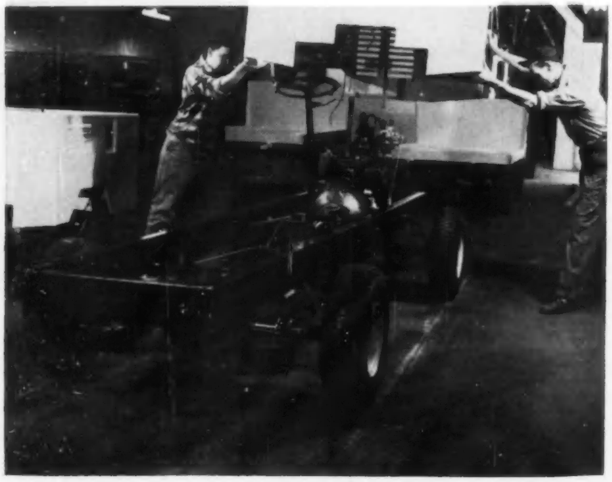

Truck frame built with USS COR-TEN Steel takes terrific torsion strains

In this truck, originally designed for use at Naval air bases, strength was the primary consideration. The builder, Kalamazoo Manufacturing Company, chose USS COR-TEN High-Strength Low-Alloy Steel for the 6-inch frame members.

The grueling acceptance test of their Model 40 (Government version of K45) Platform Truck involved hitting a series of 4-inch-high steel obstacles placed alternately on the right and left sides of the road. This racking motion set up terrific strain due to the twisting of the frame members. The test lasted 8 days, 8 hours a day. At the end of the period, there were no fractures in the COR-TEN Steel and the frame was straight and true.

Results were so good that approval was granted and Kalamazoo subsequently received a contract for 182 trucks for Naval and Marine air bases all over the world. Now Kalamazoo is making them as Model K45's for industrial use and for baggage and cargo handling at airports, with the addition of a conveyor belt.

USS COR-TEN High Strength Steel is widely used for all types of truck frames and bodies. It has a yield point 1½ times that of structural carbon steel, has 4 to 6 times the resistance to atmospheric corrosion, and offers superior resistance to abrasion and impact. What's more, paint on USS COR-TEN Steel lasts up to 50% longer than the same paint system on carbon steel.

For more information, write for a free copy of our COR-TEN Steel booklet. United States Steel, 525 William Penn Place, Pittsburgh 30, Pa.

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American Steel & Wire—Cleveland
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United States Steel Supply—Steel Service Centers
United States Steel Export Company
United States Steel

Low silhouette industrial truck built with a rugged frame of USS COR-TEN High Strength Steel and a platform of USS Multigrip Floorplate. Payload 4,000 lbs. Cargo space 45 sq. ft.

This mark tells you a product is made of modern, dependable Steel.

Transverse Properties of Steels

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differences in the transverse fatigue, impact, and tensile properties between leaded and non-leaded SAE 4140 at hardness levels of 55 and 47 Rockwell "C" are insignificant for normal application.

Where Error Comes In

The fatigue ratio of any steel is usually expressed as the ratio of endurance limit to ultimate strength. Quite often the value for ultimate strength is obtained from a hardness conversion because tensile testing is costly. However, use of the hardness relationship to determine the fatigue ratio of certain steels at ultra high strength levels is not valid if tests are made in the transverse direction because the relationship between hardness and ultimate strength does not apply to tensile tests oriented in the transverse direction. For instance, both the leaded and non-leaded SAE 4140 steels of this investigation, at a 55 Rockwell "C" hardness, had ultimate strengths of about 270,000 psi when tensile tested longitudinally, but only about 170,000 psi when tested transversely. Any fatigue ratios expressed for transverse test results can be very misleading unless tensile tests have established that the longitudinal and transverse tensile properties for the steel tested are essentially the same.

The inability of the steels investigated to develop their theoretical strength during transverse tensile tests is probably the result of inclusions elongated in the rolling direction so that they act as stress-raising notches when transverse specimens are tested. The same stress-raiser notch effect, no doubt, is responsible for the low fatigue strength values obtained and extremely poor impact test results. This conclusion is validated by data from tensile testing of longitudinal and transverse specimens with misalignment and surface flaws eliminated as factors.

Improvement in Steels

Since the effect of inclusions is the controlling factor on the transverse properties of SAE 4140, it is obvious that improvement can be attained only by use of steel which has a minimum number of harmful inclusions. Steel suppliers will not make this grade of steel on an inclusion count basis, therefore, the answer seems to lie in using a steel with inherently better transverse properties — perhaps SAE 4340 — or vacuum melted alloy steels of the SAE 4140 or 4340 grades which are now commercially available. Vacuum remelting of an SAE 4340 steel has been reported to improve the ratio of transverse to longitudinal fatigue strength

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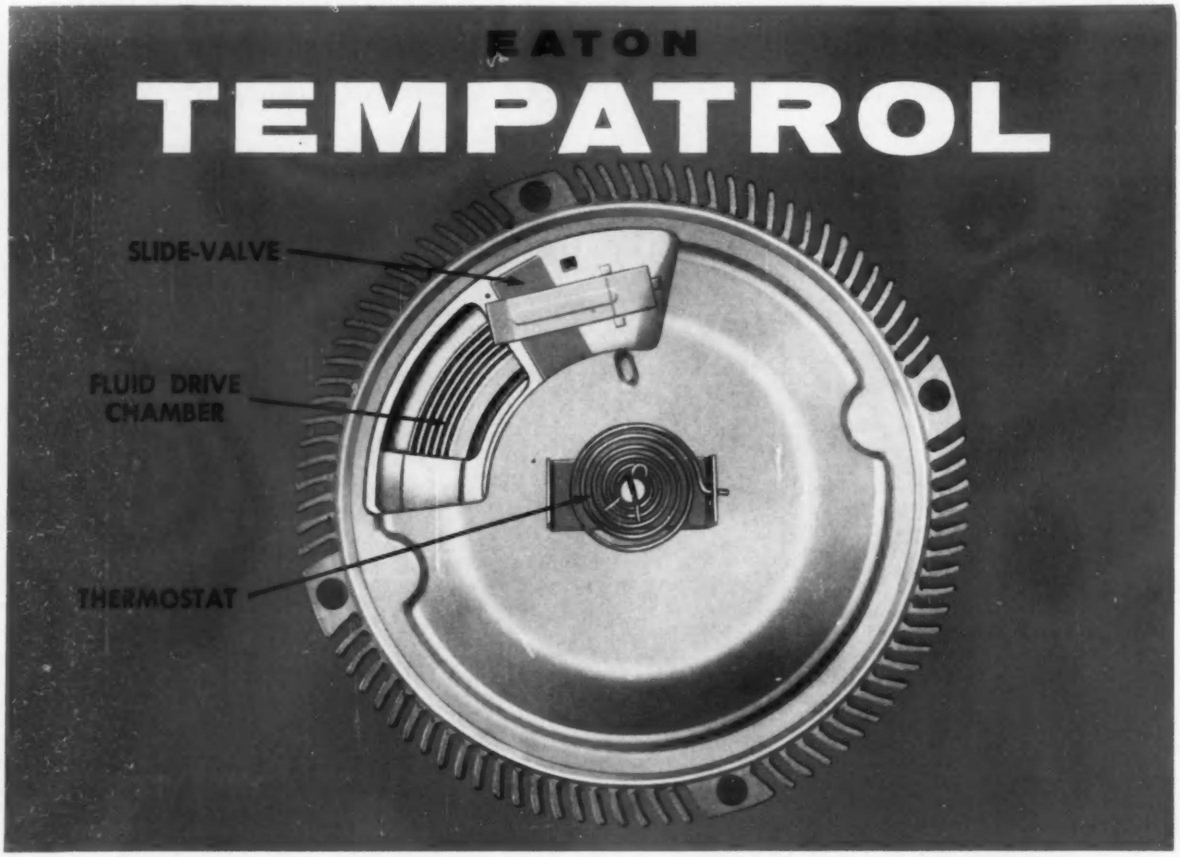
from an original value of 61% to 81%

material is stressed almost to the endurance limit, or in applications involving very high operating stresses for a lesser number of cycles.



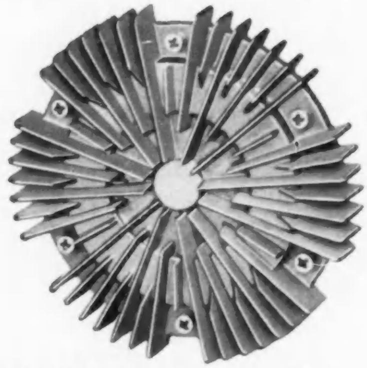
Temperature Regulated Fan Drive Increases Usable HP—Reduces Fan Noise

HOW TEMPATROL WORKS: When under-hood temperature is below the thermostat setting, slide-valve is closed; fluid-drive chamber is empty; the fan idles.

With a rise in under-hood temperature above the thermostat setting, slide-valve opens; fluid enters viscous drive chamber, increasing fan rpm to a pre-determined limit for adequate cooling.

- When cooling is not required, the thermostatic unit causes the fan to idle, thereby increasing net engine output.
- With the Tempatrol Drive engaged, adequate cooling can be achieved at low engine speeds. The top fan speed is controlled to a pre-determined maximum to avoid excessive noise.
- Operational ranges can be established to suit the requirements of each vehicle.

Eaton Tempatrol Fan Drives are readily adaptable to existing installations with only minor changes. They are now operating efficiently on leading motor vehicles. Consult with our engineers on your fan drive needs.



TORQATROL is the torque-regulated version of the Eaton Viscous Fan Drive (without thermostatic control). At low engine rpm, the fan operates at driven speeds. As engine speed increases, the viscous drive slips, limiting the maximum fan speed. Maximum fan speed limit can be raised or lowered to suit the needs of specific installations.

Tempatrol and Torqatrol are fan drive units in the family of Eaton Visco-Drives.

EATON

PUMP DIVISION
MANUFACTURING COMPANY
9771 FRENCH ROAD • DETROIT 13, MICHIGAN

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from an original value of 61% to 81% at a hardness of 41 Rockwell "C".

Data from this investigation and others and from the work of other investigators show that the drop in fatigue ratio due to the addition of lead is zero at 110,000 psi ultimate strength, 5% at 165,000 psi ultimate strength, 10.7% at 47 Rockwell "C" hardness, and 16% at 55 Rockwell "C" hardness. This 16% difference appears to be a considerable variance, but in actuality it amounts to a stress level difference of about 9000 psi — a difference significant only in applications where the material is stressed almost to the endurance limit, or in applications involving very high operating stresses for a lesser number of cycles.

Transverse impact tests showed neither leaded nor non-leaded SAE 4140 to be subject to brittle cleavage failure even at testing temperatures of − 200 F. However, both steels will fail by brittle shear fracture even at + 250 F if the principal stresses are in the transverse direction. Service and transition temperatures are not important factors for either of the steels.

■ **To Order Paper No. 220A . . .** from which material for this article was drawn, see p. 6.

Silo Launching Base Devised For Atlas ICBM

Based on paper by

R. H. THOMAS

Convair-Astronautics, General Dynamics Corp.

THE silo system is the newest operational base for the Atlas ICBM. As developed by Convair-Astronautics, the system places the missile in a vertical position in an underground cylindrical encasement where it is protected from enemy attack and ready for launching. A sectional view of the system is shown in Fig. 1.

Almost all the equipment needed to maintain and launch the missile is contained within the silo. A tunnel 50 ft long connects the silo with an underground launch control center (LCC) in which are located the launch control equipment, communications equipment, and personnel. Operating sequence calls for tanking the missile, locking the shock suspended crib to the ground, opening the silo doors at ground level, and lifting the missile to the surface for launching.

The silo is a concrete-lined hole 52 ft in diameter and 174 ft deep, topped by a reinforced concrete cap with hinged doors at ground level. Suspended within the shell of the silo is an 8-sided crib structure 154 ft high with eight levels on which are stored maintenance and launching equipment. The launch control center is also a crib structure, with one level for the crew mess and another for equipment.

Suspension Systems

Both crib and LCC suspension systems are designed to attenuate the ground shock incurred by a near-miss nuclear blast to a level tolerable to man, missile, and equipment supported

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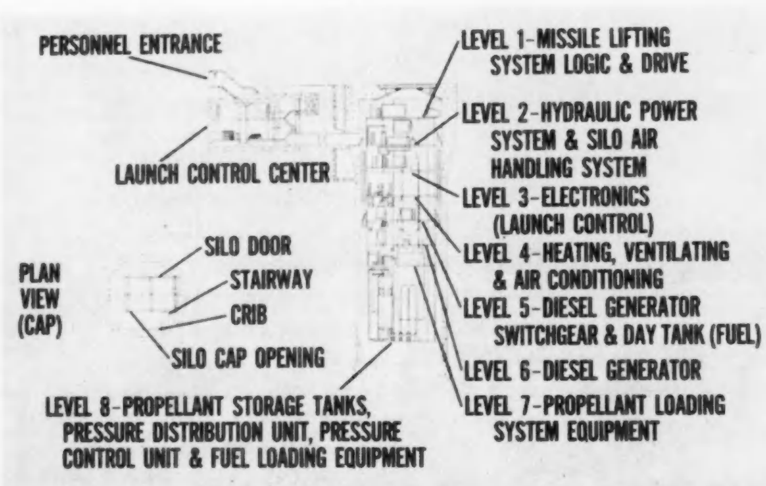

Fig. 1 — Sectional view of silo lift launcher system for Atlas ICBM. Crib structure is suspended within the concrete silo shell to attenuate ground shock from near-miss nuclear blast. Control center, located at end of 50-ft tunnel, is similarly protected.

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THE WORLD'S MOST MODERN

Rocket Fire Hazard

fuel-oxidizer fires compared with fuel- centration of peroxide vapors can re-

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on the cribs. The missile crib, weighing about 3,000,000 lb, is suspended by four pairs of struts having an equivalent pendulum length of 50 ft. Horizontal shock loads are attenuated by the pendulum effect, while vertical loads are reduced through a helical compression spring system with an approximate spring rate of 189,000 lb per in.

The LCC suspension system also operates on the pendulum system except that vertical shock load is attenuated by a pneumatic spring system. The smaller mass of the LCC crib permits use of a low-pressure pneumatic system with a good deal of flexibility.

Launcher platform guide rails, drive mechanism, counterweight, drive cables, and locking system comprise the lift or elevator system. The fully loaded missile is elevated approximately 123 ft in 60 sec, using a peak power demand of 225 kw. Lowering time and elevation time of an empty missile is about 10 min because of the nature of the over counterweighted system and the power restriction.

The missile is raised and lowered on a launcher platform — a steel structure 17 ft square and 45 ft high. A flame deflector is built into the upper section of the structure to divert the missile exhaust horizontally just above the silo cap.

■ **To Order Paper No. 241A . . .** from which material for this article was drawn, see p. 6.

Fire Hazard Climbs With Rocket Vehicles

Based on paper by
KENNETH P. LOPP
North American Aviation, Inc.

EXOTIC fuels and fluids, aerodynamic heating at high-speed, high-altitude operation, and higher performance systems, associated with rocket-powered vehicles such as the X-15, have multiplied the risk of fire. And they have changed designing for fire safety from an art to a scientific undertaking.

Combustible Fluids

Hypergolic propellants, those fuel-oxidizer combinations which ignite spontaneously upon contact without the presence of an ignition source, pose an extremely hazardous condition if there is leakage, since fire is a certainty and the isolation of ignition sources is ineffective. Nonhypergolic propellants, on the other hand, require an ignition source for initiation. They are less likely to cause fire if they leak, but sufficient quantities can

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Rocket Fire Hazard

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accumulate to cause damaging explosions if ignition occurs. Pyrophoric propellants ignite spontaneously on contact with air, requiring no ignition source. In a sense, they are hypergolic with air, or an oxidizer; however, all hypergolic propellants are not necessarily pyrophoric. Pyrophoric fuels are especially hazardous on leaking because air cannot be entirely eliminated from surrounding areas.

The higher reaction temperatures of fuel-oxidizer fires compared with fuel-air fires is another factor to consider. Ordinary fuel-air fires encountered in earlier vehicles resulted in flame temperatures of 2000 F; fuel-oxidizer fires in rocket vehicles can produce flame temperatures of 5000 F.

Contamination is a hazard with some fuels. Hydrogen peroxide, for example, dissociates into oxygen and superheated steam (about 1300 F) when contaminated by certain impurities. This high temperature might overheat structure or equipment, or it might serve as the igniter of combustible fluids in the presence of the liberated oxygen. A relatively low per cent concentration of peroxide vapors can react with explosive force upon contamination.

Ignition Sources

Aerodynamic heating of equipment or structure introduces an auto-ignition potential and there is also the problem of the potential for high energy ignition of combustibles. Common sources of the latter are electrical and electronic equipment. Electrical arcing or sparking by such equipment during normal operation or as a result of failure can readily ignite combustibles. Another source is produced by static electrical buildup caused by high-speed flight or high flow rates in vehicle fluid systems. Subsequent discharge or arc-over can be of sufficient energy to ignite combustibles.

Several incidents or fire or explosion have been traced to ignition sources generated by high energy radar beams. Metallic debris lying in the beam path is known to absorb energy resulting in arc-over to adjacent structure or other metal objects thereby causing ignition of combustibles. It is suspected that certain types of fluids may be sufficiently heated by absorption of energy from radar beams to cause ignition.

In space vehicles, effects upon electrical and electronic equipment are anticipated as a result of dense spacial regions of ultraviolet and cosmic radiation, ozone, and ionized gas concentrations.

To Order Paper No. 242D . . .
from which material for this article was drawn, see p. 6.

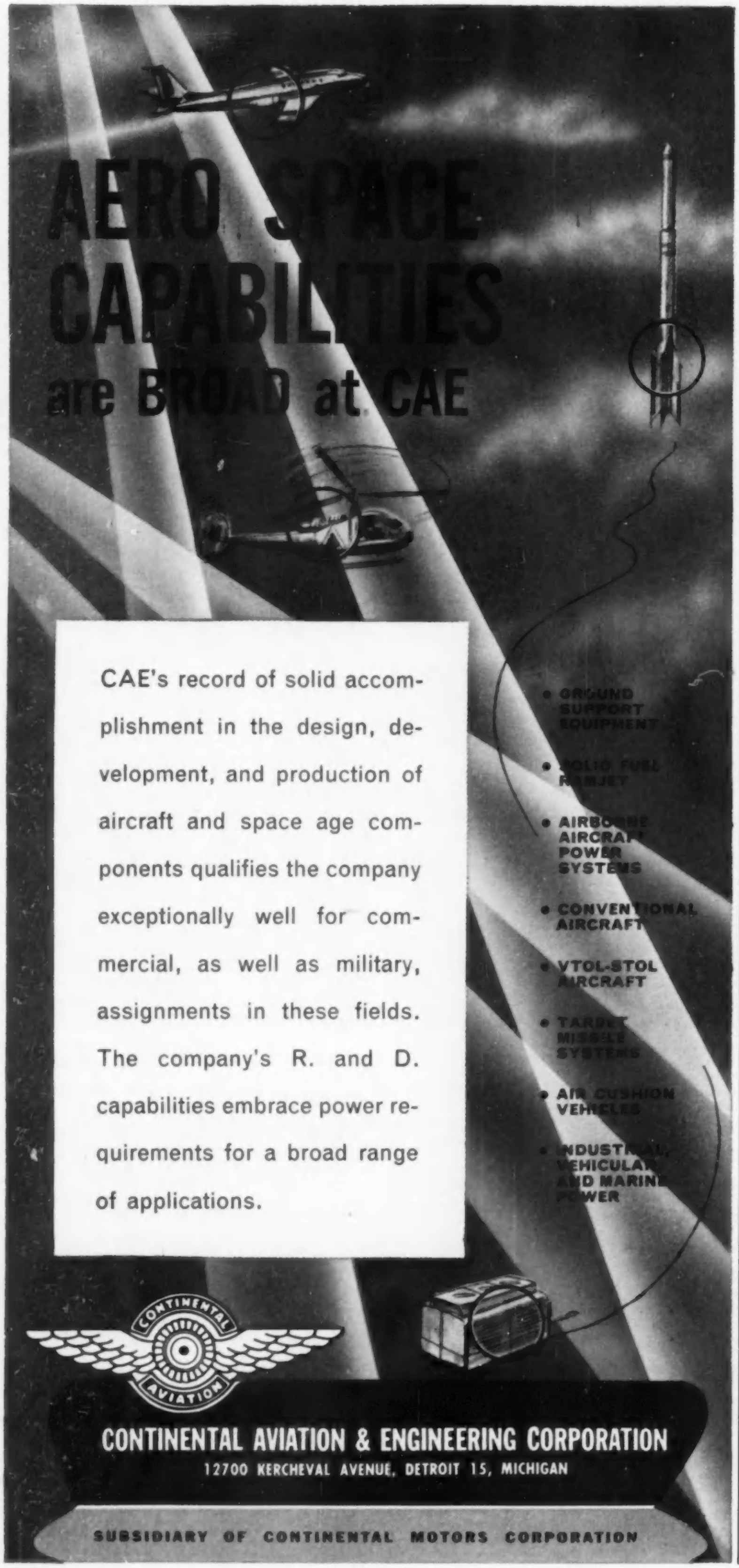
Cracking, H Stripping Help Bring Autoignition

Based on paper by
C. E. Welling, G. C. Hall, and J. S. Stepanski
Scientific Laboratory, Ford Motor Co.

FUEL CRACKING and hydrogen stripping may be contributing to autoignition, it is suspected as a result of recent Ford studies on concurrent pyrolytic and oxidative reaction mechanisms in the precombustion of hydrocarbons.

In these researches, the cracking and hydrogen stripping reactions in the precombustion of a paraffin fuel were indicated to afford sources of H atoms, HO_2 free radicals, and perhaps indirectly of OH free radicals. And since one or more of these species are generally believed to be the active agents in autoignition, it is reasonable to assume that cracking and hydrogen stripping may contribute to autoignition.

To Order Paper No. 201B . . .
from which material for this article was drawn, see p. 6.



New Members Qualified

These applicants qualified for admission to the Society between September 10, 1960 and October 10, 1960. Grades of membership are: (M) Member; (A) Associate; (J) Junior.

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Central Illinois Section: William P. Edwards (M), Donald Allen Holtkamp (A), Alan H. Olinger (J), Weldon Leon Phelps (J), Frank H. Riddle (J), Willis O. Schueth (M).

Chicago Section: William Harrison Bateman, Jr. (M), Ugo G. Frisoni (A), William E. Gaskill (J), Edward Carl Grahn (J), Lee Carlton Hoppe (J), Frank William Jenks (M), William C. Keeran (M), Franklyn J. Lammers (M), A. W. Lawson (M), James H. Lowe (M), Brian Keith Neil (J), Evan G. Olson (A), Gerald E. Ritchey (J), Jerome Conrad Rosenwald (J), Paul A. Sanders (M), Charles B. Tracy (M).

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You've never seen a motor like this!

New

NEW ONE-OF-A-KIND MICROGRAPH
draws pictures for bearing research

This greatly magnified stylus is drawing a picture of the microscopic imperfections in a bearing raceway . . . measuring each one to within a few millionths of an inch. The picture-on-tape which comes out of this specially modified micrological instrument is an important tool in BCA's research on ball bearing performance.

This is just one of the precision instruments in the Temperature-Humidity-Controlled Instrumentation Room which is the center of BCA research on bearings. The result of this program is revealed in on-the-job performance of BCA bearings. They roll dependably under heavy loads and all kinds of adverse conditions.

New testing facilities at the BCA laboratories also include specially designed equipment, often identical with equipment in customers' plants. Here, BCA bearings are tested to exceed customer specifications *under the exact operating conditions experienced by the customer!*

BCA ball bearings are standard original equipment . . . replacement, too . . . for nearly every kind of industry. For example, automotive, earth moving, agricultural and machine tools. The wide line of ball bearing sizes and types, plus BCA's research and extensive new testing facilities, pays off for bearing users. Consider the performance record of BCA ball bearings the next time you purchase or specify bearings. For more information, or for assistance with bearings problems, contact Bearings Company of America, Division of Federal-Mogul-Bower Bearings, Inc., Lancaster, Pa.

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NEW FLUID STEERING

New Members Qualified

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nor (M), Jack Ogden (M), Russell De-Witt O'Neal (M), Robert F. Pecha (M), Lawrence Glen Peck (J), Orville E. Phelps (M), John Popovich (M), Charles H. Raches, Jr. (J), Norman Douglas Rice (M), Edmund F. Sarosy (M), Samuel Herbert Schram (M), Donald L. Shepard (M), Donald J. Smith (A), Fernley G. Smith (M), Jerome T. Smutek (J), Glenn Walter Steinman (M), Arthur J. Stock (M), Richard L. Suter (M), Frank George Tenkel (J), Terry A. Tetens (J), James A. Turck (A), David N. Viger (A), Jon Stephen Wilson (J).

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Oregon Section: Russell Lowell Madsen (A).

Philadelphia Section: Leonard J. Grecco (M), Gerald George Kroninger (J), Harry William Lenczyk, Jr. (J), Francis P. Lentine (J), James M. Lewis (J), Charles Carroll McClelland (J), Dineschandra S. Shah (J), Richard Joseph Teti (J), Lowell Eugene Van Zandt (A), Gilbert Zweig (J).

Pittsburgh Section: James W. Butler, Jr. (M), David Michael Capone (J), Joseph M. Krippendorf (A), Gerald R. Petrey (A).

Rockford-Beloit Section: Paul F. Marshall (J), Romas Balys Spokas (J), Richard C. Zimmer (J).

Salt Lake Group: Gerald Leon Coakley (J), William Clyde Neal (J).

San Diego Section: Robert Bowen (A), Billy Don Sevier (J).

South Texas Group: Paul T. Whitmore (M).

Southern California Section: Samuel D. Gray, Jr. (M), Alfred Herman Greiert, Jr. (A), Frank O. Jappel, Jr. (A), William F. Knabb (M), Joseph Patrick Mulligan (J), Ernest Lloyd Phelps (J), Bernard J. Sadoff, Jr. (J), John R. Singer (A).

Southern New England Section: Douglas Gale Culy (J), George Richard Krohne (A), Furman Hovey Martin, III (J), C. William Steele, Jr. (J).

Syracuse Section: Clifford G. Banks (A), James Owen Tennies (J).

continued on p. 150

New Members Qualified

continued from p. 149

Texas Section: Wendyl Bruce Baker (J).

Washington Section: Joseph P. Smith, Jr. (M).

Western Michigan Section: Jack Ernest Larson (J), Floyd K. McGahan (A).

Wichita Section: John Francis Melaugh, Jr. (J).

Williamsport Section: Ralph E. Thomas (A).

Outside Section Territory: William Ronald Brookes (J), John Joseph Fox (A), Harold G. Friday (A), John McNulty Horne (J), Robert E. Huhnke (A), Bert James Minshall (J), S. H. Mitchell (M), Roger Frank Ort (M), Robert W. Rue (M), Joe Earl Taucher (A), Robert Joe Wicke (J), John Roger Wilson (J), Joseph W. Wiswall (J).

Foreign: Wong King Chen (M), Malaya; Robert Jones (M), East Africa; M. S. Seetharaman (J), Germany.

Applications Received

The applications for membership received between September 10, 1960 and October 10, 1960 are listed below.

Alberta Group: Eric Brown, Roy Andreas Flarup, Vernon George Seadon, Ernest Paul Villet

Baltimore Section: Abraham G. Emanuel

British Columbia Section: Donald Arthur Brundrett

Buffalo Section: A. Richard Antinarelli, Michale Francis Doyle, William P. Neumeister, Paul Channing Stimson

Central Illinois Section: Raymond Edwin Diefendorf, Ernest Henry Griéme, Theodore Mathew Kero, Benny Ballheimer, Warren Francis Budds, Frederic P. Buttke, Donald Elmer Lull, James O. Machlan, Don R. McGuire, Edwin L. Riedesel, Raymond Engene Schuler, Ray Shipman, William Porter Sly, Morris Alfred Swanson

Chicago Section: R. J. Aberle, William R. Bogett, Thomas E. Brady, Gordon E. Cole, Rodney R. Erickson, Clifford W. Hager, Jerrold A. Isaacson, James S. Jackson, Dan Bruce McVickar, Charles Michael Neugebauer, Alan Joseph Peterson, Philip Otho Schafer, F. W. Recktenwald, Robert F. Wenger

Cincinnati Section: William Henry Gwynn, W. William Harting

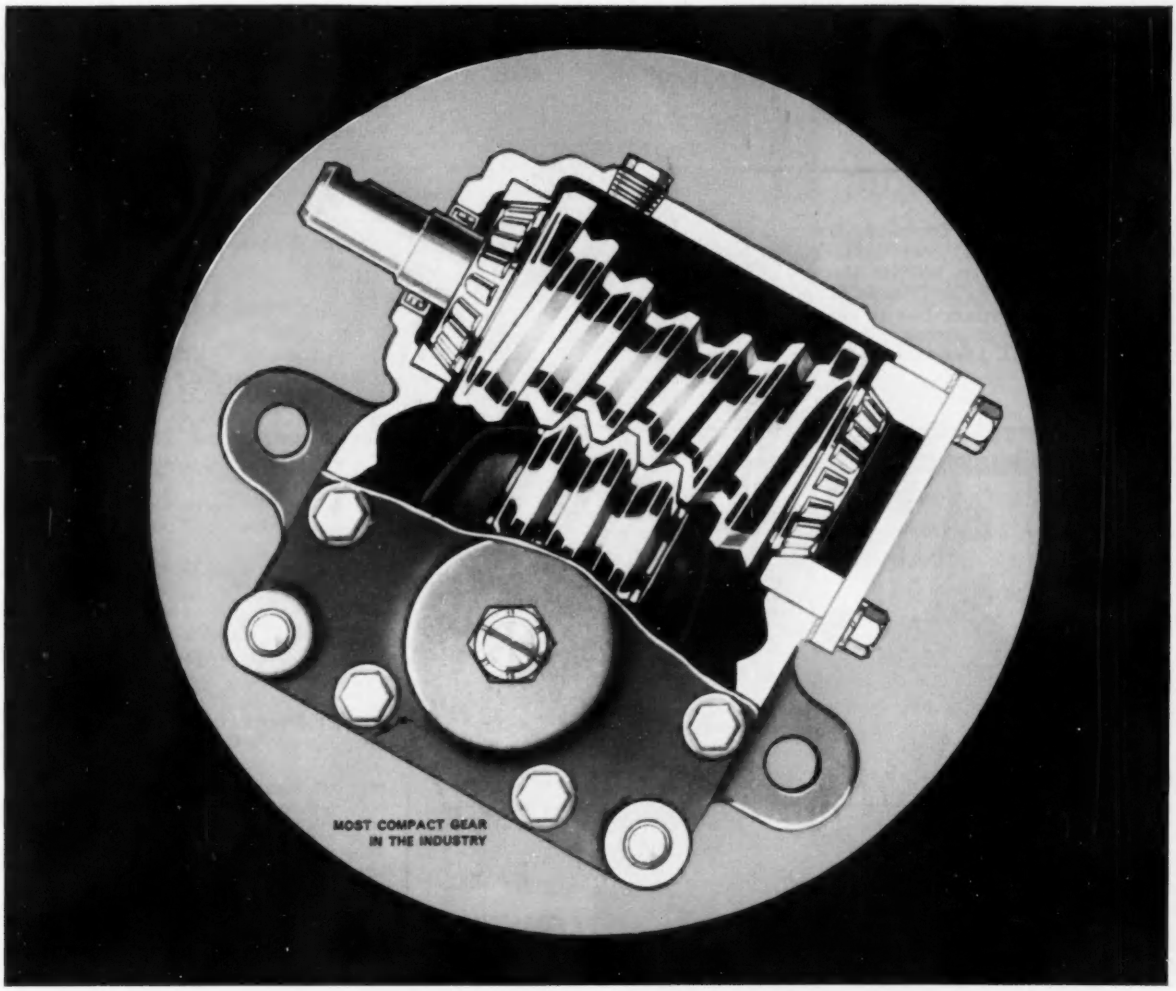
Cleveland Section: John W. Brodhacker, Lawrence Hume Chenault, Gary Michael Gron, Robert P. Klenner, Samuel Adam Kuzoff, Steven Ottokar Luzsicza, Francis H. Pascoe, John R. Perkins, William Rodgers, Donlad E. Wall, Charles Francis Walton

Dayton Section: Merle Julius Athmer

Detroit Section: Michael M. Bahn, Edward Richard Betz, James Garland Brown, C. Borden Chase, Dewitt W.

continued on p. 152

Bach, Frederick W. Bloom, Stanley E. Chocholek, Donald A. Darnell, Roscoe



SMOOTHER STEERED WHEN GEMMER GEARED

Gemmer steering is engineered for strength and ease of operation . . . built for lifetime service and minimum maintenance. Customer pleasing features of the 7D gear are:

Compactness . . . High numerical ratio of 28:1 . . . An efficient, rugged gear . . . Simplicity of installation and adjustment . . . Available in either malleable or aluminum housings

For *your* steering needs, Gemmer offers you experience as old as the industry.

Gemmer steering
SINCE 1906

ROSS GEAR AND TOOL COMPANY, INC.

Gemmer Division, Detroit, Michigan • Ross Division, Lafayette, Indiana

Applications Received

continued from p. 150

Cooper, Robert Allen Darovich, Dan Griffin, Jr., Michael Murray Hentgen, Eugene Bernon Hilker, William David Horn, Peter Edgar Hoyal, Tadeusz Idzikowski, Iver S. Johnson, Iver S. Johnson, Oscar John Kubicek, Maurice Leising, Stephen J. Linsenmeyer, Louie Jackson Lipp, Gilbert Thomas Lyon, Daniel Henry Moir, Franklin Delano Obermeyer, Edward B. Ray, John Leo Rex, Jr., Richard Arlan Rider, Kenneth L. Romig, Jr., Walter T. Szymanski, Paul C. Timmerman, Edward S. Wellock, Howard O. Wold , Charles F. Bach, Frederick W. Bloom, Stanley E. Chocholek, Donald A. Darnell, Roscoe W. Davis, Jr., Richard Marvin DuBois, John R. Elwell, Earl R. Fiene, Richard B. Gould, Robert C. Johns, Ronald J. Narlock, William M. Spaller, Thomas Edward Stuck, James Arthur Weller

Fort Wayne Section: Arthur B. Bok, Vernon A. Johnson, Herbert Garth McKen, Robert E. Smart, William A. Williams

Hawaii Section: Bruce Noboru Takamine

Indiana Section: Eugene K. Buchholz, James W. Crooks, Carlton Lee Duncan, Charles W. Edson, Dan C. Fuller, Floyd Arnold Kunce, Floyd L. Smith, Tracy H. Wolfe

Kansas City Section: Wilbur Gene Ragains, Allen Eugene Tucker

Metropolitan Section: Michael Bojcik, Jr., Frederick W. Clemens, James J. Harford, David Jacobs, Gottlieb Koenig, Gerald Kopelman, John George Markel, Edward J. Roche, Robert Scott Stahr

Mid-Continent Section: Billey Paul Hensely, Allyn G. Warkentin

Mid-Michigan Section: Frank Donald Baber, Gerald C. Born, Robert Russell Haist, Earl L. Helmers, Dan R. Kimberlin, Ronald Lloyd Kyle, James E. Marshall, Marshall W. Miller, Joseph A. Stearns, Robert Carl Stempel, Ralph Dudley Whittier

Milwaukee Section: Lucian M. Darin, William James Downie, George W. Gross, Donald H. Mathes, Milan Petrovich, Robert S. Reaves, David Eugene Schuh, James Walsh

Mohawk-Hudson Section: John C. Krambuhl

Montreal Section: Robert Richard Bryce, Roy Randall, Joseph Camille Rivet, Robert Harry Wright

New England Section: Allen H. Maynard, Charles H. Murphy, Robert Joseph Raymond

Northern California Section: Earle E. Dodd, Kenneth Le Roy Eckhardt, Jack C. Gunther, Jr., Owen R. Knop, Bertram Jonathan Leigh, Peter Francis Melia, Fernando Jose Munoz, James Russell Roger, Loren N. Williams

Northwest Section: Ronald Gene Lenz, Laird W. McKee

Ontario Section: Clayton G. Busher, Gunnar E. Liepins, Donald Charles Buckley, Merlyn Eugene Cranston, Walter A. Hadden, Donald Alexander Peterson, Richard Frederick Robert-

continued on p. 155

SILICONE NEWS from Dow Corning

Applications Received

continued from p. 152

son, Donald Anthony Rossetti, Reginald B. Spencer, Robert Weir

Oregon Section: Martin Philip Davis, Louise Sullivan

Philadelphia Section: Joseph Vito Cresko, Jerome Eugene Daniels, David Jerome Dwyer, William Robert Kane, Donald Leroy E. Terry, Earl R. Walton, Jr.

Pittsburgh Section: Walter E. Gregg

St. Louis Section: John L. Niebrzydoski, Walter Franklin Powell

Salt Lake Group: James D. Calvin, James McCoy Workman

San Diego Section: Theodore C. Tyce

Southern Texas Group: Paul Neilson Howell

Southern California Section: Philip Mitchell Beck, Elliott David Brown, Harold D. Daigh, James L. Edman, Leland Price English, Ben G. Franco, Jr., Iwao Ishimizu, Allen Kanov, Michael William Kiekak, William D. McGarrity, Frederick P. Morgan, William Rust, Thomas M. Wambaugh, Robert A. Wilson

Southern New England Section: Joseph Max Friedberg, Paul Rudolf Rey, Vincent J. Sansevero, Jr.

Spokane-Intermountain Section: Harley E. Stober

Texas Section: James Clifton Blackmon, J. A. Kerr, Donald J. Weber

Twin City Section: Clifford C. Bigley

Virginia Section: Harvey Burwell Bennett

Western Michigan Section: Paul F. Bergmann, Jr.

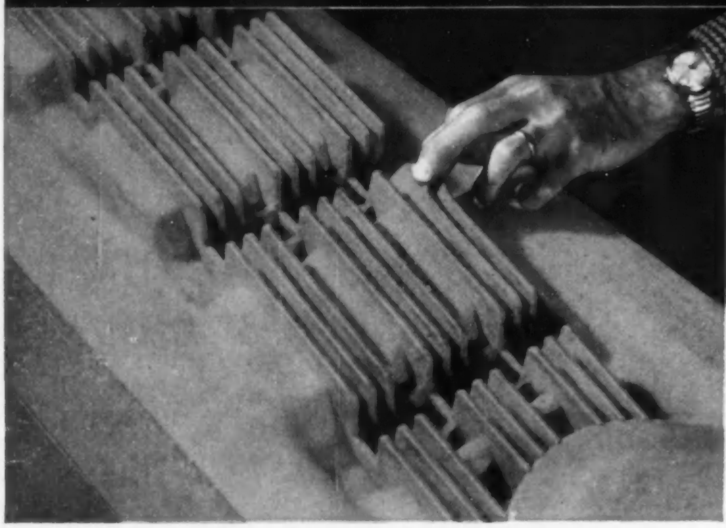
Williamsport Group: Milton Ronald Lederer

Outside Section Territory: John Craton Armold, Don D. Cummins, Nicholas Tony Frangias, Carl George Fritz, Harold Magne Fuglvog, Raymond Filmore Gillen, John Joseph Kass, Russell S. Kenerson, Alfred L. Neuhoff, Frank Nowak, Jr., Marshall D. Sawdey, Harold M. Schneider, Jr., Keith Louis Spencer

Foreign: Vishan Chandra Agarwal, India; William Doyne Chadwick, Australia; Hans Hessler, Brazil; Jesus Torres Moncayo, Mexico; Adul Pinsuvana, India; Roy C. Rainsford, Australia; George Verghese, India; Kasi I. Krishnan, India; Somasundram Kannan, India; Kasi Gopala Mohandas, India

SILICONE NEWS from Dow Corning

Perfect Prototypes . . . Fast!



Silastic RTV puts designs into 3-D easily, quickly, economically

Turning intricate designs into prototypes and finished parts is greatly speeded and simplified with Silastic® RTV, the versatile fluid silicone rubber that vulcanizes at room temperature. Prototypes made with this new engineering "tool" are exact reproductions of the original patterns.

How it's done.

Here's how one automotive parts supplier uses Silastic RTV in producing small quantities of experimental grilles. The first step, illustrated at top, is a precise wooden pattern of the design. Next, plaster is poured over this pattern to form the first mold. Then, Silastic RTV takes over. This fluid rubber flows easily into the plaster mold, conforming to the finest detail, then cures at room temperature to form a firm-but-flexible *exact* duplicate of the original wood pattern.

Now, a second plaster mold is made from the silicone rubber duplicate. Alloys are poured into this plaster mold to form finished metallic prototype parts.

Why bother with Silastic RTV and the extra step? Several good reasons. First, Silastic RTV releases the plaster mold quickly, smoothly — something that's difficult with wood originals. Second, by "standing-in" for the original wood pattern, Silastic RTV protects that costly pattern from harm. Another important benefit is durability. The silicone rubber duplicate can be used over and over again!

Silastic RTV resists temperatures up to 500 F, enabling you to make prototypes directly from the RTV mold with plastics. Examples: trim parts, instrument panels and many other parts important to overall design and beauty.

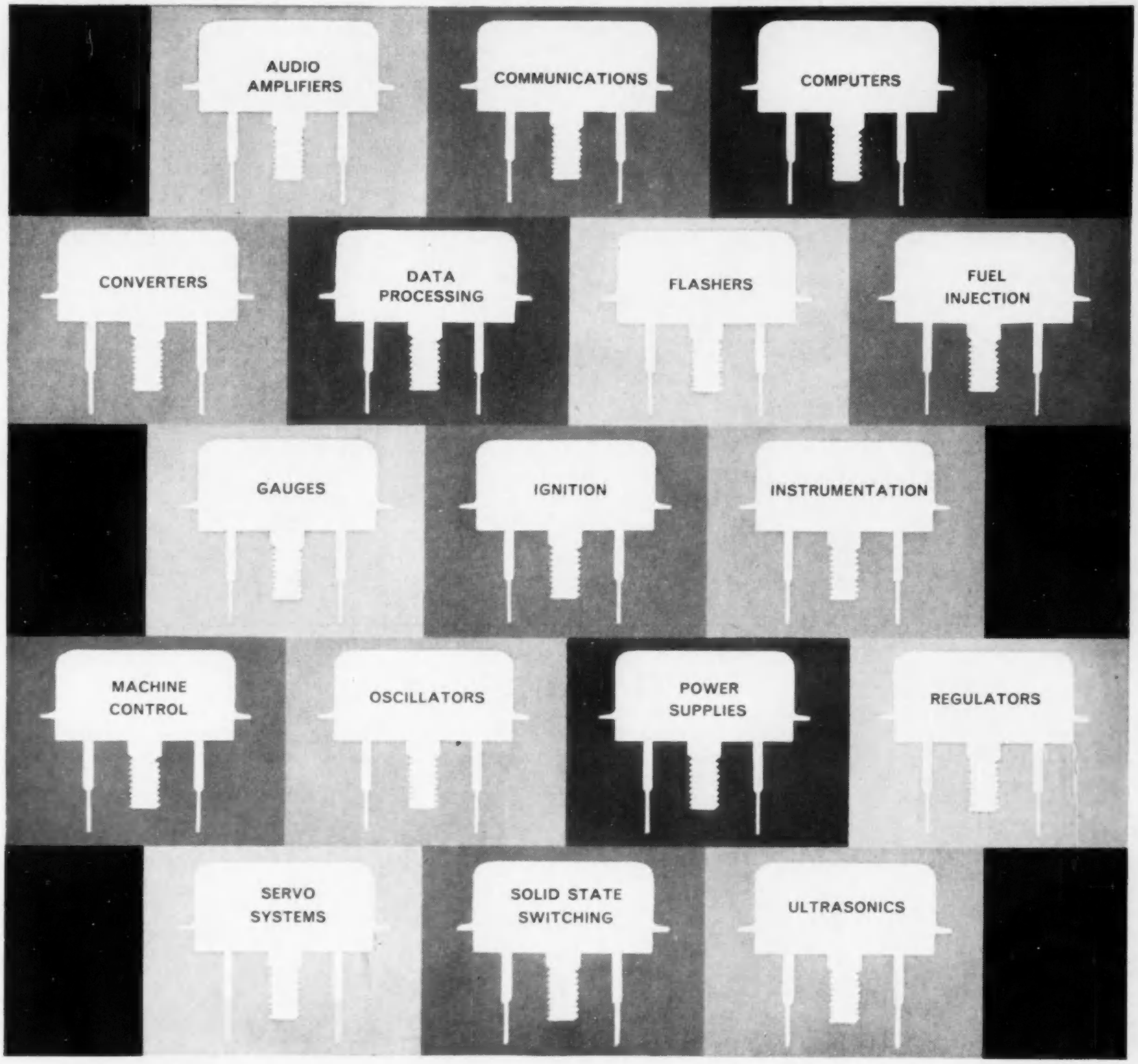
Every day more and more engineers turn to Silastic RTV for help in cutting costs, in reducing time requirements, in pretesting of new designs. How can this versatile material best serve *you*? Investigate today.

For detailed information on Silastic RTV, contact the Dow Corning office nearest you, or write Dept. 8011.

first in silicones

Dow Corning CORPORATION
MIDLAND, MICHIGAN

ATLANTA BOSTON CHICAGO CLEVELAND DALLAS LOS ANGELES NEW YORK WASHINGTON, D. C.


AUDIO AMPLIFIERS
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DELCO
2N174

lower

your

cost of

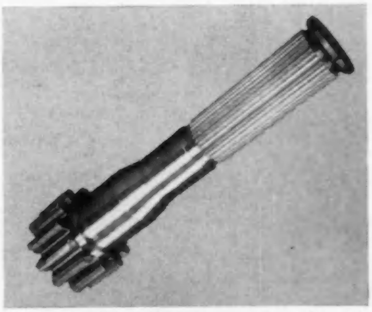
possession

2 Nickel Alloy Steels provide

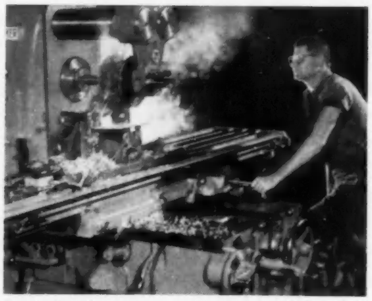
A simple plan to save money by Steel Standardization

Holding the Shippingport nuclear reactor's top head to its shell section is a job performed by 42 studs of AISI 4340. This through-hardening nickel steel provides essential strength and toughness.

Power-packed tractor features unique drive mechanism composed of regular gear transmission, clutch and torque converter with lock-up, utilizes nickel alloy steels in more than 25 components.



Swing shaft for giant power shovel made from 4340 nickel alloy steel for strength and toughness to take shock-loading in stride. This 9"-diameter, 36"-long shaft of 4340 nickel alloy steel transmits tremendous torque to a giant ring gear.



Sustained accuracy is assured in this heavy-duty milling machine by spindle and gear components of AISI 4340 and 4620 nickel alloy steels. They provide needed strength and wear resistance.

How many types of steel do you normally use?

Chances are you should examine your needs for alloy steel to see how many can be satisfied with just two types, 4340 through-hardening and 4620 carburizing.

You are likely to find that most of your engineering requirements can be met with confidence when you *standardize* on these two general-purpose steels.

You simplify inventory and materials-handling. You save money in purchasing and production, too.

- 4340 stands alone among medium-carbon steels in its ability to provide maximum strength, ductility, toughness and resistance to fatigue in parts of medium to heavy section.
- 4620 is a carburizing type that has consistently proved itself the ideal steel for a wide variety of carburized parts. 4620 is easy to carburize and has a minimum tendency toward distortion in heat treatment.

Available From Steel Service Centers

Both 4340 and 4620 are stocked by Steel Service Centers from coast to coast, ready for off-the-shelf delivery in a variety of sizes.

When you have carried your standardization plan as far as you can and you still have specialized needs to fill, the right nickel steels are available to give you extra performance or even lower costs.

For a buyer's guide to Steel Service Centers that stock 4340 and 4620, and other nickel-containing grades, simply write Inco.

INCO TRADE MARK

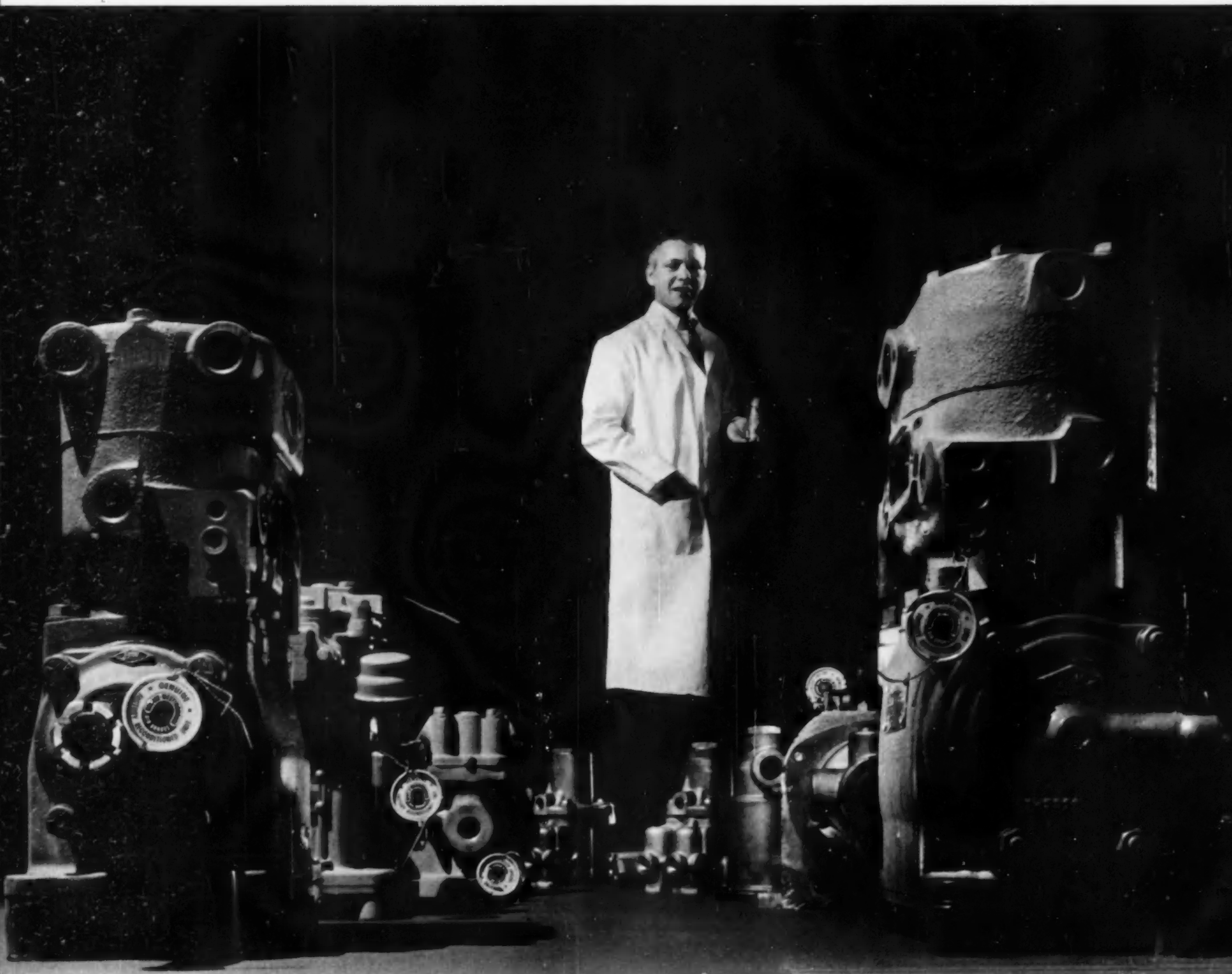
THE INTERNATIONAL NICKEL COMPANY, INC.
67 Wall Street, New York 5, N. Y.

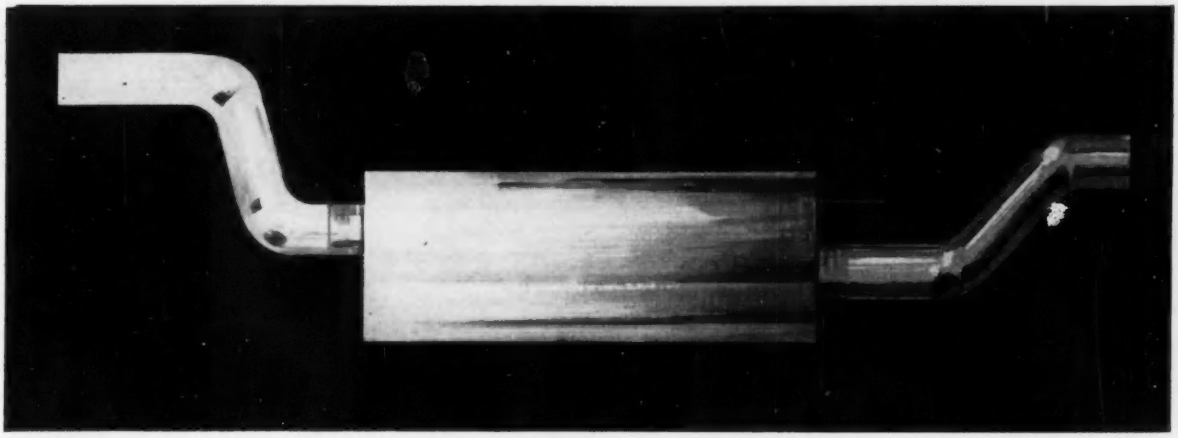
STEEL SERVICE CENTER INSTITUTE

"SYSTEMATIC" ECONOMY. . . regular piggy-banking assures it . . . and when it comes to braking so does Bendix-Westinghouse! For when you specify Bendix-Westinghouse Air Brakes, you get the kind of long-range economy, performance and dependability that only a complete air brake system with all components designed and engineered to work together can give you. That's why most of the nation's leading fleet operators as well as vehicle manufacturers choose complete air brake systems by Bendix-Westinghouse. When it comes to new vehicles, be sure to get the safest, surest, most economical stops available—Air Brakes by Bendix-Westinghouse.

SPECIFY COMPLETE AIR BRAKE SYSTEMS BY *Bendix-Westinghouse*

BENDIX-WESTINGHOUSE AIR BRAKES





Protect Exhaust Systems From End to End

with Armco ALUMINIZED STEEL

New steels are born at Armco

No two areas of an auto exhaust system suffer exactly the same kind of destructive attack. In some locations, varying combinations of heat and corrosive condensate chew at exhaust system parts. Table 1 gives an idea of the destructive constituents of typical exhaust condensates. Laboratory tests indicate an average PH for similar condensates of about 2.7. In other areas, heat alone is the enemy.

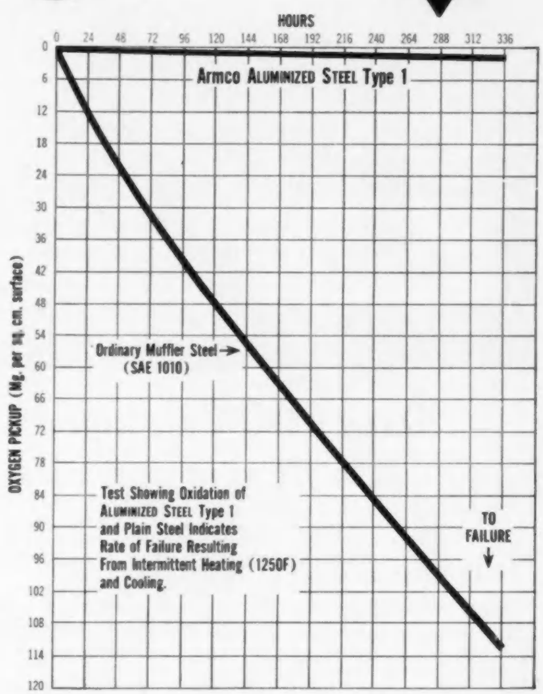
Wherever the attack, however, Armco ALUMINIZED STEEL Type 1 provides longer service life. Against deadly combinations of heat and corrosion ALUMINIZED STEEL stands up longer than any metal in its price class. Where high temperature is the major culprit, ALUMINIZED STEEL provides many times the resistance of carbon steel, as evidenced by Graph A.

In short, Armco ALUMINIZED STEEL Type 1 in exhaust system parts saves car owners money and trouble—gives auto manufacturers an important sales feature. For more information on this durable hot-dip aluminum coated steel, write Armco Steel Corporation, 2770 Curtis Street, Middletown, Ohio.

TABLE 1

AN ANALYSIS OF EXHAUST CONDENSATES

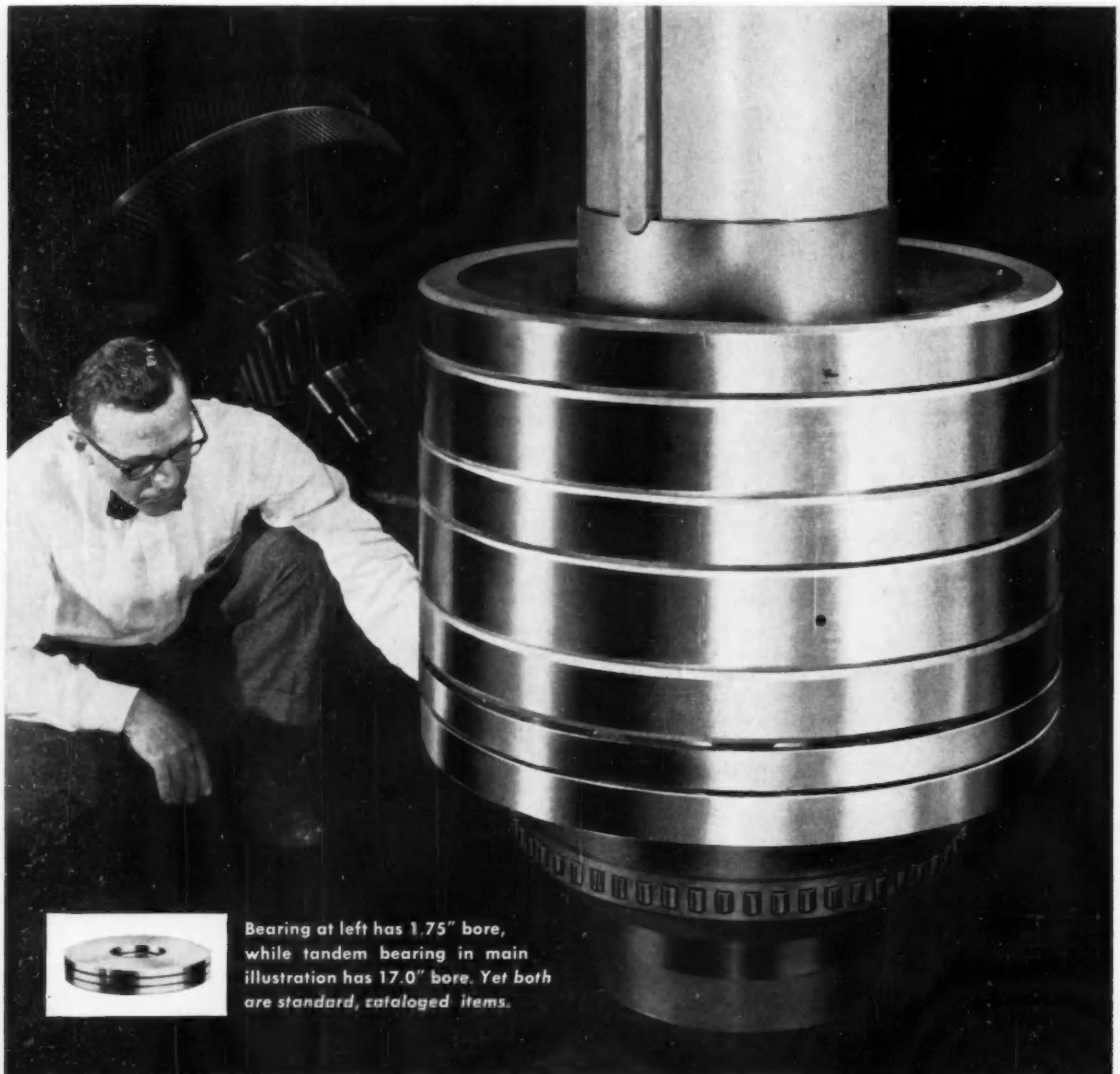
Constituent	Concentration—ppm
Sulfates **(SO_4)**	**690**
Chlorides **(C1)**	**520**
Bromides **(Br)**	**370**
Lead **(Pb)**	**9**

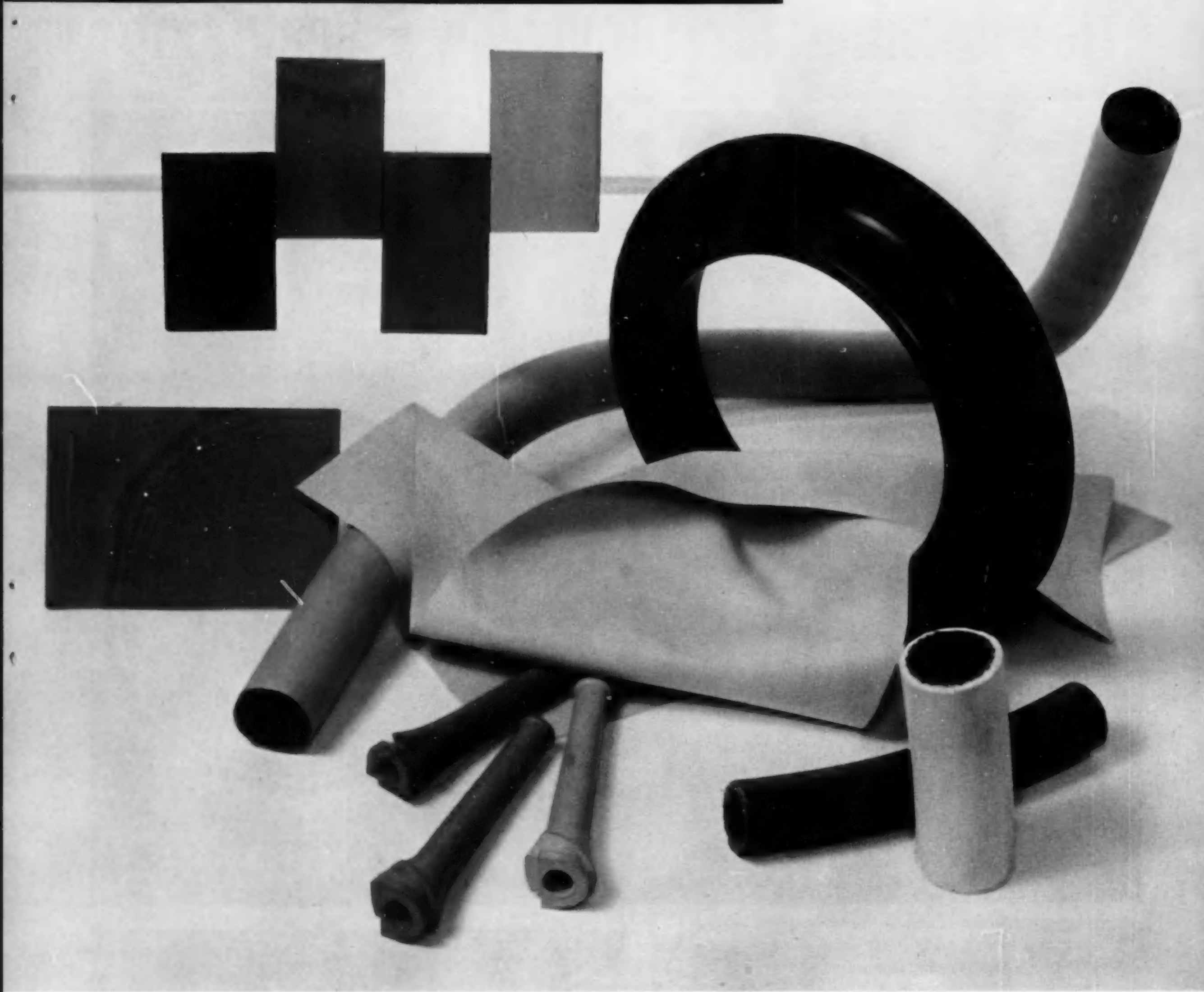


ARMCO STEEL

Armco Division • Sheffield Division • The National Supply Company • Armco Drainage & Metal Products, Inc. • The Armco International Corporation • Union Wire Rope Corporation

THE OIL-RESISTANT, OZONE-RESISTANT NITRILE RUBBER


Bearing at left has 1.75" bore, while tandem bearing in main illustration has 17.0" bore. Yet both are standard, cataloged items.



BUILD IN STRENGTH AND LIGHT WEIGHT WITH N-A-X FINEGRAIN STEEL

In a wide variety of applications N-A-X FINEGRAIN—a low-alloy high-strength steel—can bring important weight savings without sacrifice of strength and safety. When you design N-A-X FINEGRAIN into your product, you get *all* these advantages:

- It is 50% stronger than mild carbon steel.
- Polishes to a high luster at minimum cost.
- Can be cold formed readily into difficult stampings without surface disturbances.
- Is stable against aging.
- Has high fatigue life with great toughness.
- Has greater resistance to wear and abrasion.
- Is readily welded by any process.
- Offers greater paint adhesion.
- Has twice the resistance of carbon structural steel to normal atmospheric corrosion.*

*Where greater resistance to extreme atmospheric corrosion is important, our N-A-X HIGH-TENSILE is recommended.

Sound like something for you? A thoroughly competent metallurgical service organization is available to work with you on any application problem you may have. Write, wire or phone Product Development Department, Great Lakes Steel Corporation, Detroit 29, Michigan.

N·A·X
FINEGRAIN STEEL

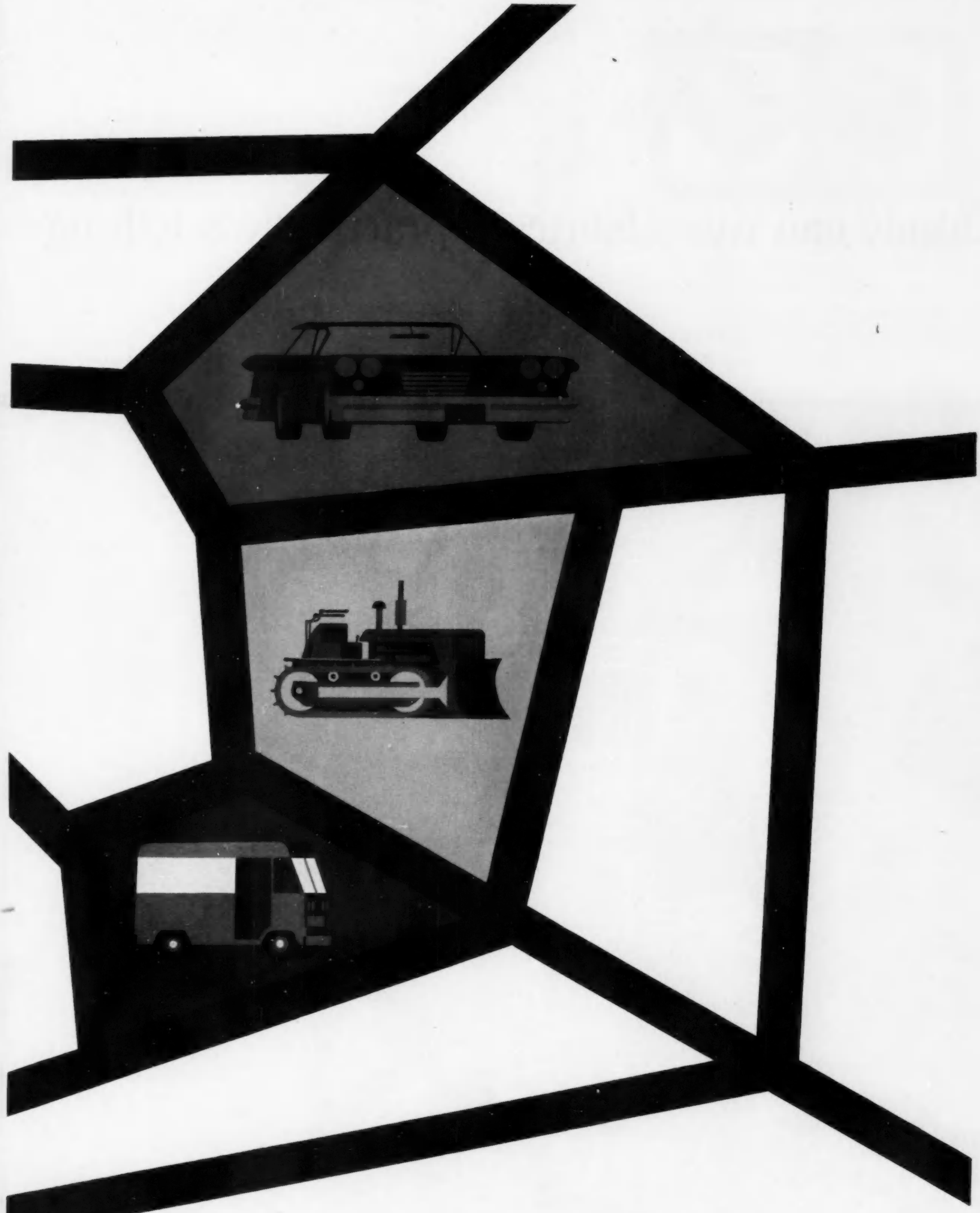
A PRODUCT OF

GREAT LAKES STEEL

Detroit 29, Michigan

Steel

Look for the STEELMARK on the products you buy; place it on the products you sell.



The American Automotive Industry—the world's

greatest enterprise—depends on tire accomplishments

The American Automotive Industry—the world's

Thousands of extra vehicle miles

AIR CONTROL Since 1844

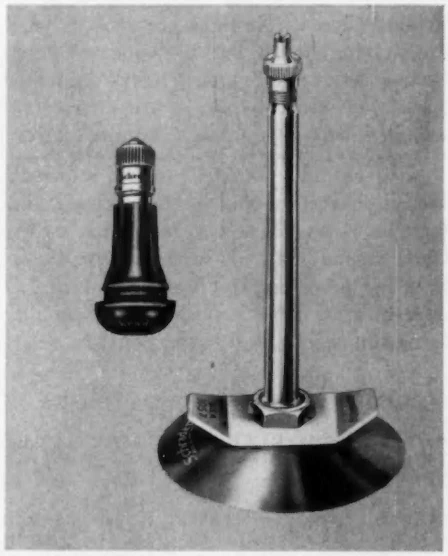
Schrader's famous tire valve operating principle is the Ace of Standardization for today's longer lasting, better performing tires.

greatest enterprise—depends on tire accomplishments

in every tire and tire valve... through rubber chemistry

The Automotive Industry consumes 60% of the world's rubber, chiefly in the form of tires*. These tires are amazing . . . they give superior performance . . . they are economical. Reduction of aging, deterioration and wear of tire valve rubber is a result of specialized companies working together. Only by utilizing combined skills, experience and facilities can tire valves be produced to match each others' performance where it counts . . . on the vehicle itself. Count on Schrader's finest rubber chemistry techniques and facilities to supply quality tire valves for any tire-equipped vehicle anywhere in the world.

*Encyclopedia Britannica



BONDING RUBBER to metal valve inserts requires multiple matching: a metal alloy, to an etching technique, to a rubber compound . . . plus, for tubes, the additional requirements of rubber-to-rubber bond.

NO ONE RUBBER meets *all* requirements of *all* valves. Schrader adds special ingredients to natural rubber, SBR, butyl, neoprene, and other kinds to achieve a balanced combination of desired properties for each specialized valve—tube—tire.

RESULT: The tire on every vehicle rolling gives the driver maximum safety and economy. Schrader rubber chemistry makes a major contribution to world-wide vehicle performance.

A. SCHRADER'S SON • BROOKLYN 38, N. Y.
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Schrader®
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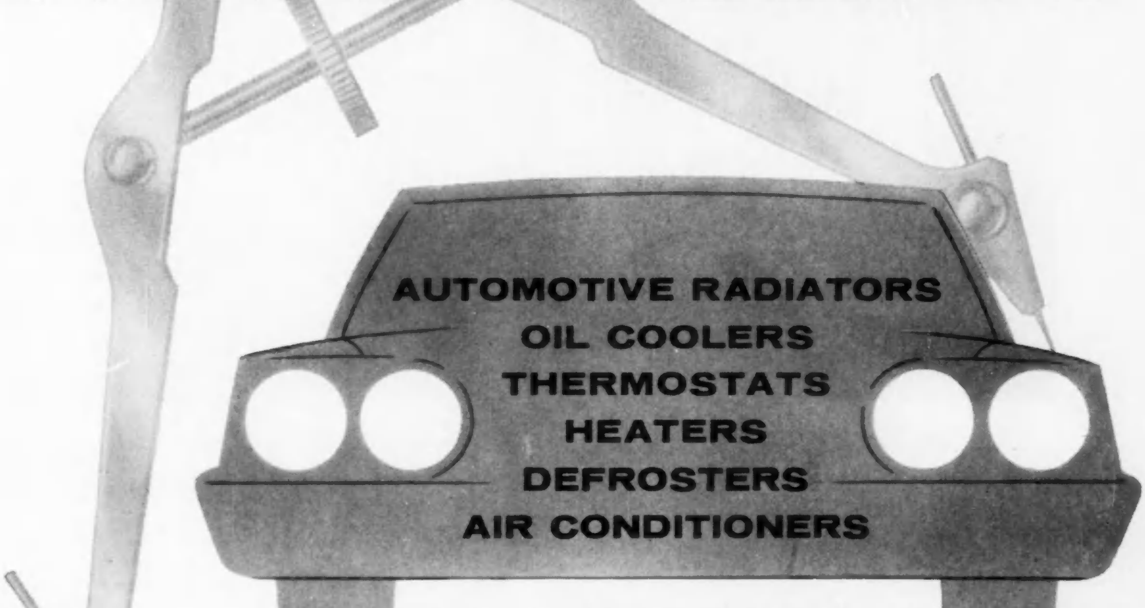
FIRST NAME IN TIRE VALVES

FOR ORIGINAL EQUIPMENT AND REPLACEMENT

Through General Motors Leadership...

GM HARRISON OFFERS THE MOST EXACTING DEGREE OF RELIABILITY



For over fifty years, Harrison Radiator Division of General Motors has pioneered, perfected and produced reliable heat transfer equipment for the automotive industry. Reliability is a reality at Harrison. Reliability from idea to end-use, based on scientific controls that cover every phase of product development and manufacturing . . . proved by millions of motoring miles. Your company and your customers can benefit from this Harrison heat-transfer experience in the automotive field. So, if you have problems with temperatures ...passenger comfort or vehicle efficiency...look to the leader in reliability. Look to Harrison for the answers.

GM HARRISON

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AUTOMOTIVE RADIATORS • OIL COOLERS • THERMOSTATS • AIR CONDITIONERS • HEATERS • DEFROSTERS

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INTERNATIONAL S·A·E AUTOMOTIVE ENGINEERING CONGRESS · EXPOSITION

In one place . . . at one time . . .
a world preview of engineering advances
in GROUND and FLIGHT vehicles

COBO HALL • DETROIT • JANUARY 9-13 • 1961

Plan to Attend the SAE International Congress and Exposition of Automotive Engineering . . .

. . . in Detroit's New Convention Center — Cobo Hall.

WHO IS COMING

15 thousand men charged with design and manufacture or selection and maintenance of passenger cars, trucks, buses, farm tractors, construction equipment, transport aircraft, missiles, space vehicles — and their components and materials.

WHAT'S IN IT FOR YOU

If you're one of them, set aside January 9–13 to acquaint yourself with current innovations in all fields of automotive engineering as they will be portrayed at the Congress and Exposition. Allow yourself time to tour the Exposition, look up your old friends and make new ones, and to participate in plant trips . . . SAE Annual Banquet . . . technical committee meetings . . . Fathers and Sons Night . . . annual business session of the Society . . . luncheon honoring foreign guests . . . meetings of any Activity Committees and administrative committees of which you are a member.

NONMEMBERS WELCOME

Nonmembers of the Society can attend the Congress and enjoy many of the advantages of membership temporarily by paying the registration fee of $3.00 per day or $12.50 for the five days. No fee for SAE members, of course, nor for members of the armed forces and other government employees, students, and faculty members.

WHY SO BIG

The opening of Detroit's huge new convention center, Cobo Hall, enables SAE to make its 1961 Annual Meeting the largest gathering of engineers concerned with self-propelling ("automotive" that is) vehicles of all kinds ever held under one roof. The expanded 1961 SAE Annual Meeting is being called the SAE International Congress and Exposition of Automotive Engineering.

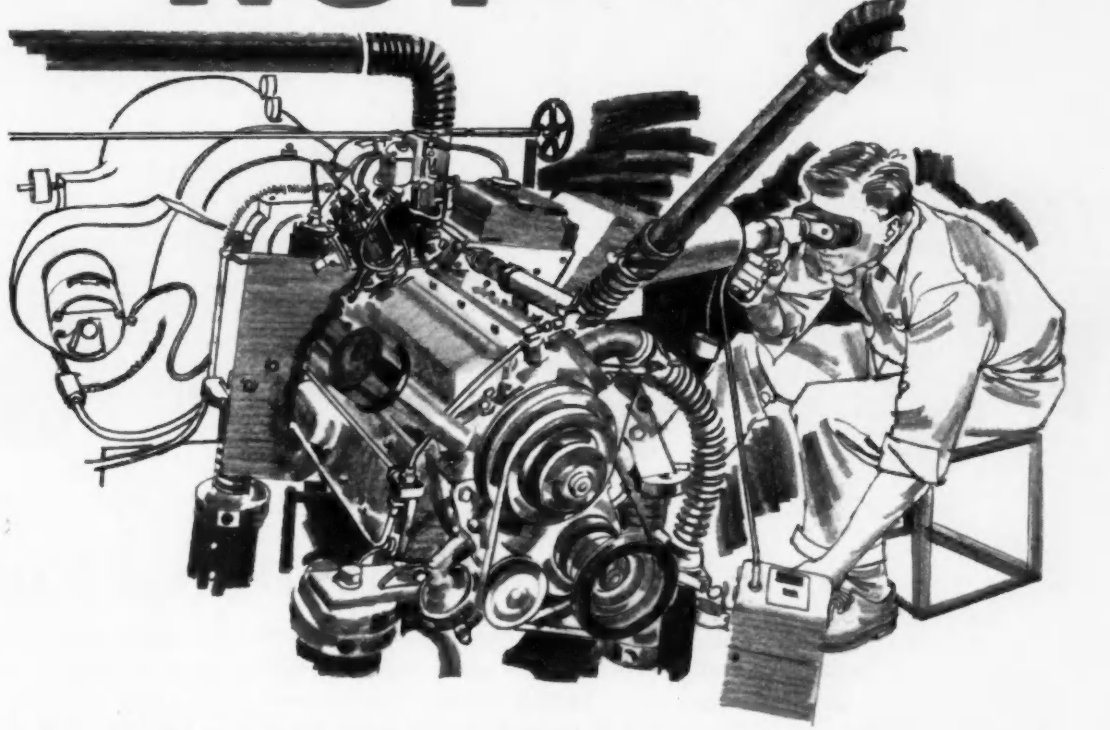
LARGEST DISPLAY OF ITS KIND

Inspect the products and developments of the more than 300 exhibitors who will augment the technical program with the latest in manufacturing techniques, systems and components, materials and powerplants. In the 4½ acre display area, you'll have the opportunity to question the technical representatives manning these booths, and to get the answers to *your* problems.

FOR INFORMATION CONTACT:

MEETINGS DEPARTMENT
SOCIETY OF AUTOMOTIVE ENGINEERS
485 LEXINGTON AVE.
NEW YORK 17, N.Y.

How HOT are Your Valves?



Eaton Technique Determines Valve Operating Temperatures—Permits Accurate Selection of Valve Materials

1. The valve to be studied is duplicated in a special steel which has a known temperature response in terms of hardness.

2. The temperature-responsive valve is run for a brief period in a test engine, under the operating conditions which are of interest to the engine builder or user.

3. After removal from the engine, the valve is sectioned in the metallurgical laboratory, and checked for hardness at all strategic points. From these hardness indications the corresponding temperatures at which the valve was running can be accurately computed.

This Eaton technique permits the selection of suitable valve materials and designs to meet the requirements of each specific engine. Eaton Valves—made from materials selected or developed by means of the above technique—are setting outstanding performance records.

EATON

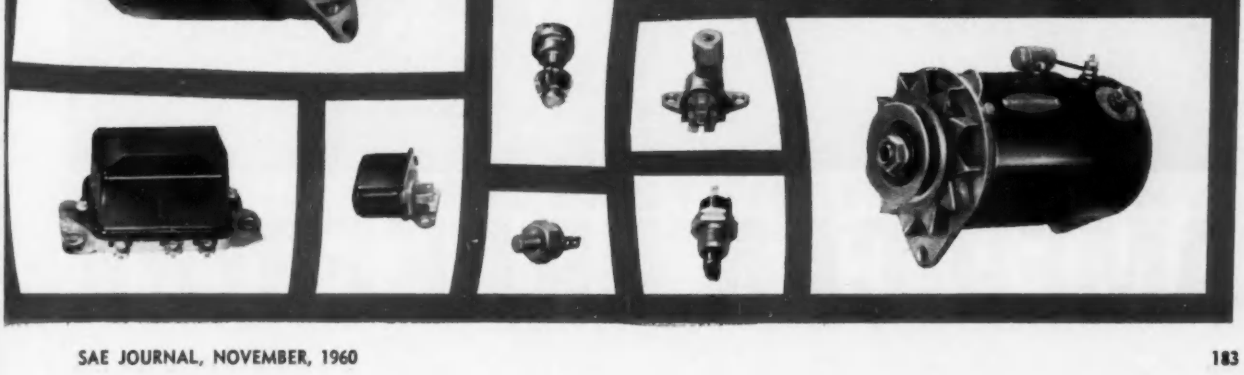
The Eaton Valve Division is fully equipped to supply all of the services necessary for valve temperature determination, including the making of the temperature sensing valves. Call on us.

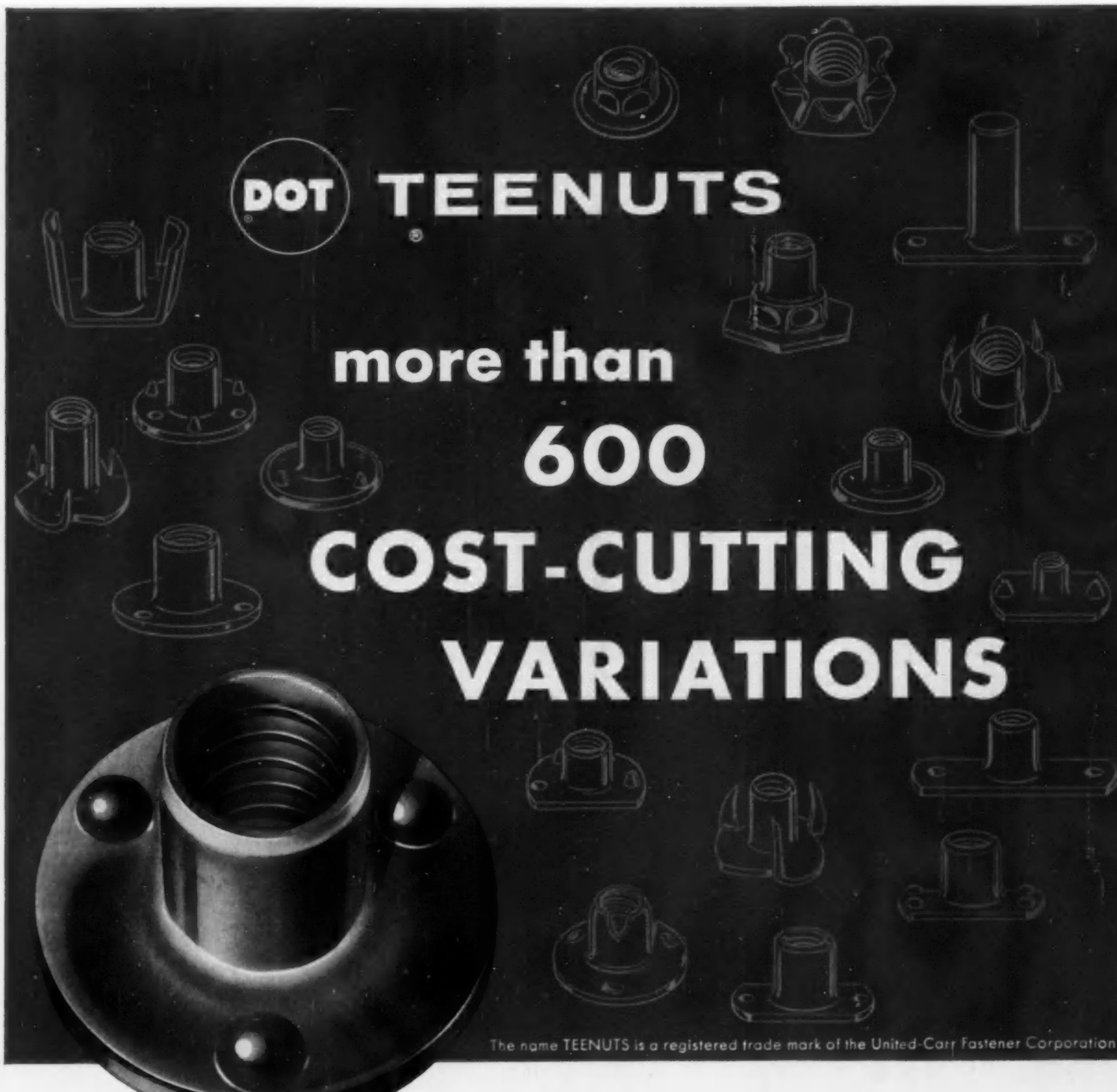
EATON

VALVE DIVISION
MANUFACTURING COMPANY
BATTLE CREEK, MICHIGAN

1961

1961





Since the first TEENUT was developed by Carr Fastener in 1927, more than 600 different modifications of this extremely versatile device have been designed and manufactured in true, mass-production quantities.

By combining nut and washer in one solid unit, the DOT TEENUT offers exceptional strength and security and eliminates the need for tapping. Its flanged base can be formed with welding bosses for attachment to sheet or solid metal structures . . . with prongs for wood . . . or with any number of different special bases for particular applications. DOT TEENUTS can be made in heat and corrosion-resistant materials and they can be provided with moisture-seals and vibration-proof, self-locking barrels.

Once mounted, the DOT TEENUT stays put and can't be lost or mislaid . . . an advantage at any time and a necessity where blind fastening is required.

Wide experience in the proper application of DOT TEENUTS and a multitude of other special-purpose fasteners enables your DOT field representative to provide prompt and effective solutions to a tremendous variety of fastening problems. Where special design work is needed, he can bring you the services of a design-engineering group unequalled in its field.

The DOT TEENUT catalog is an invaluable reference . . . yours on request.

MAKERS OF ★ DOT ★ FASTENERS

CARR FASTENER COMPANY

Division of United-Carr Fastener Corporation, Cambridge 42, Mass.

Offices In:

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25
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SPECIAL NOTICE—
TO ALL PAST, PRESENT AND PROSPECTIVE EXHIBITORS IN
SAE's INTERNATIONAL EXPOSITION OF AUTOMOTIVE ENGINEERING

PLAN NOW FOR YOUR MOVE TO DETROIT'S COBO HALL IN 1961...

Here's What Ward's Automotive Reports Have to Say About the Move:

SAE Annual Convention Booked at Detroit's Cobo Hall, 1961-1965

The Society of Automotive Engineers has engaged Detroit's Cobo Hall for its annual conventions from 1961 through 1965.

The move will give SAE the opportunity of sponsoring what could be the most prominent automotive engineering display in the country and would undoubtedly add authority to Detroit's standing as the motor capital of the U.S. and heart of the industry.

The modern Detroit Civic Center site, currently under construction in the city's bustling downtown waterfont section, is scheduled for opening in August, 1960.

January Dates Set

The SAE business dates firmed up at this time for the 1961-1965 conventions are: 1961 – Jan. 9-13; 1962 – Jan. 8-12; 1963 – Jan. 14-18; 1964 – Jan. 13-17; 1965 – Jan. 11-15.

Cobo Hall will provide the SAE sessions with 400,000 sq. ft. of exhibit space contrasted to just over 10,000 sq. ft. at Detroit's Sheraton-Cadillac Hotel, where the January meeting was held this year.

Membership Swells

It would not be unlikely that SAE will rent exhibit space during its convention to various trades connected with the auto industry for individual expositions.

Textile manufacturers and leather firms would be able to set up equipment to detail their fabric-making processes; rubber makers could show how a tire is made; similar exhibits could be allotted to the replacement parts business; car manufacturers, themselves, might devise cutout working models of engines or even entire automobiles or trucks in simulated motion.

The whole SAE affair could, in fact, house minor conventions for just about every engineering trade allied with the automotive and accessories business.

SAE's expanding membership has been a primary factor in the society's search for larger convention quarters. As of Jan. 1, there were 23,000 members, with the count swelling every month.

Excerpt—Wards Automotive Reports
March 30, 1959

Why Not an Automotive Engineering World's Fair at Detroit's Cobo Hall

Such a structure as Cobo Hall, situated as it is in the manufacturing heart of the auto industry, could be a perfect place for a gigantic technical exhibition—practically a world's fair of automotive engineering – sponsored by the Society of Automotive Engineers.

What a progressive industrial advance would be made by SAE's promotion of a colossal automotive engineering convention-exhibit, particularly with such a valuable location as Cobo Hall available!

Suppliers Could Participate

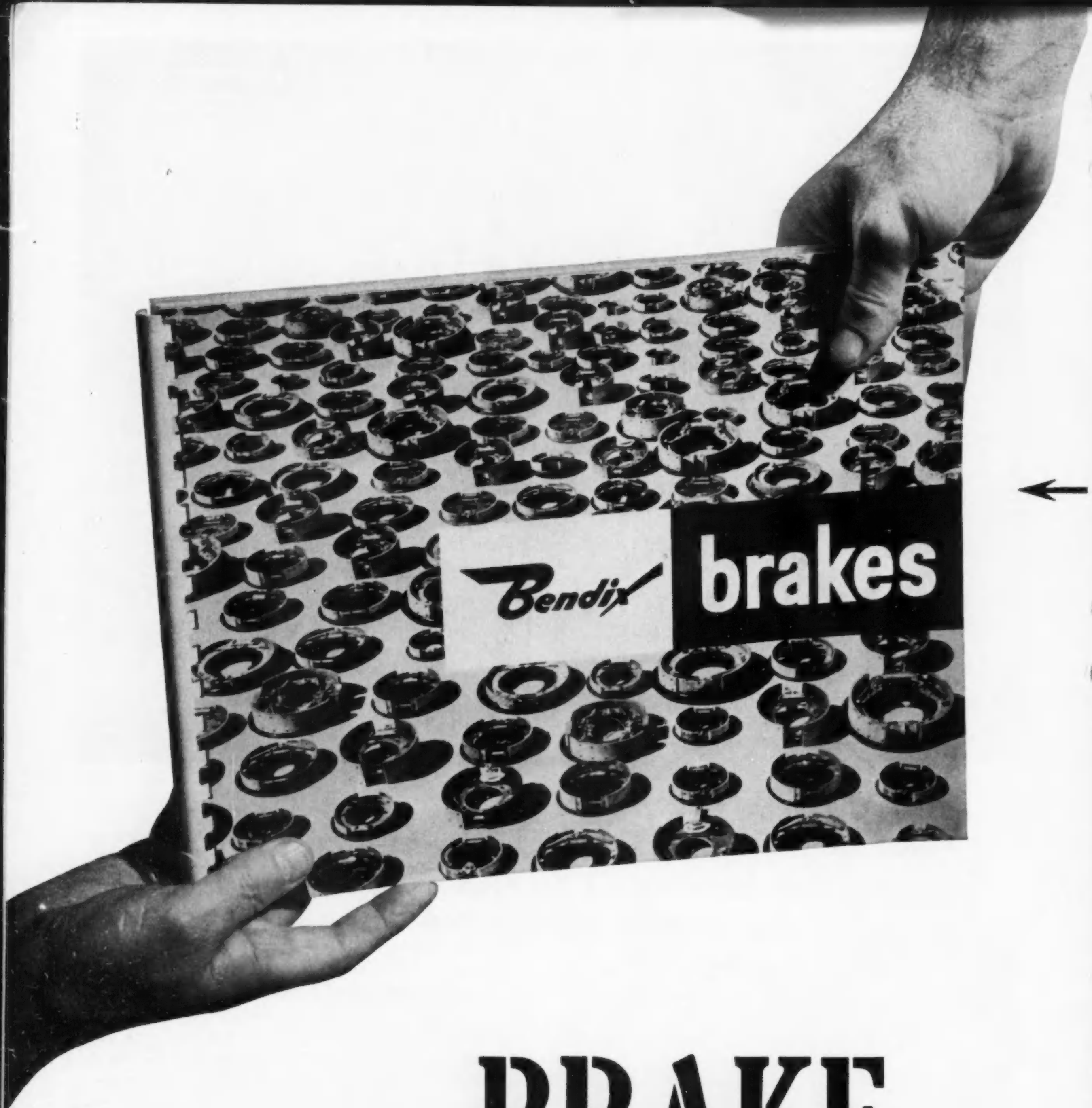
Parts makers, rubber and tire firms, textiles and leather companies, the metals trades – all of these groups and everyone else with a piece of automotive equipment to show or sell could be provided with the space sufficient to properly present and if necessary, demonstrate his advanced design product.

Cobo Hall's foundations are strong enough to hold heavy equipment such as huge body element stamping presses and various types of rugged metal working machines. The machinery could turn out stampings or tools right in the exhibit area.

If SAE could come up with such a spectacle it would certainly sweep crowds into Detroit, throngs from various areas of industry and business. The affair could, in fact, house minor conventions for every technical trade allied with the auto and accessories field.

SAE selected Cobo Hall for 1961 and subsequent conventions not only because of its vast exhibit area but for other accommodations as well, including several meeting rooms that seat over 500 persons and a banquet and adjacent room that can hold and serve over 5,000. *Excerpt*—Ward's Automotive Reports

HERE'S WHAT YOUR 1961 AUTOMOTIVE MARKET PLACE LOOKS LIKE:



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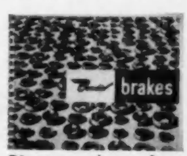

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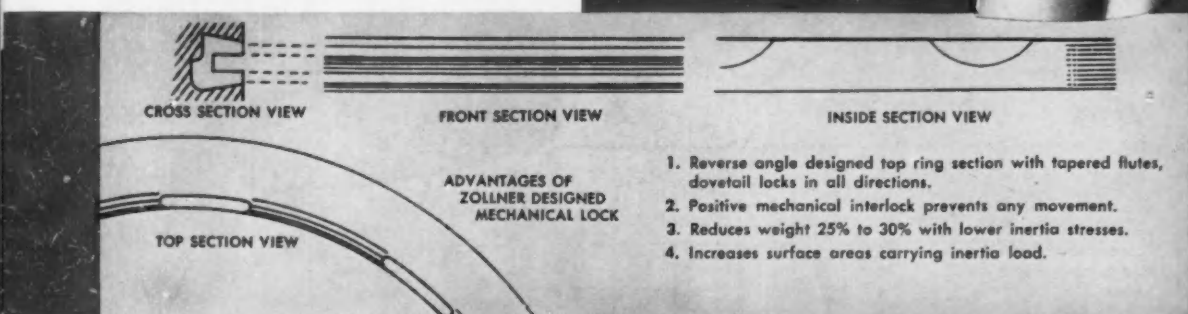

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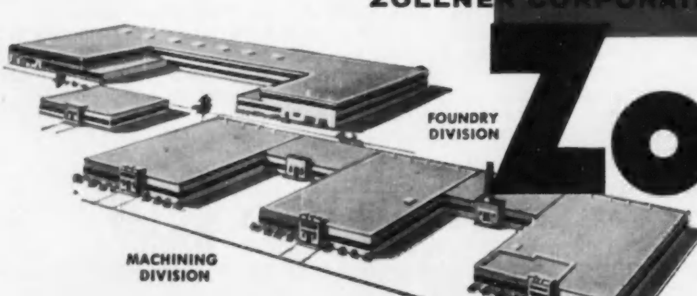

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